OCTOBER, 1949
Vol. 5 - - - - No. 5

Lubrication ENGINEERING

PHILADELPHIA
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JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

Trabon

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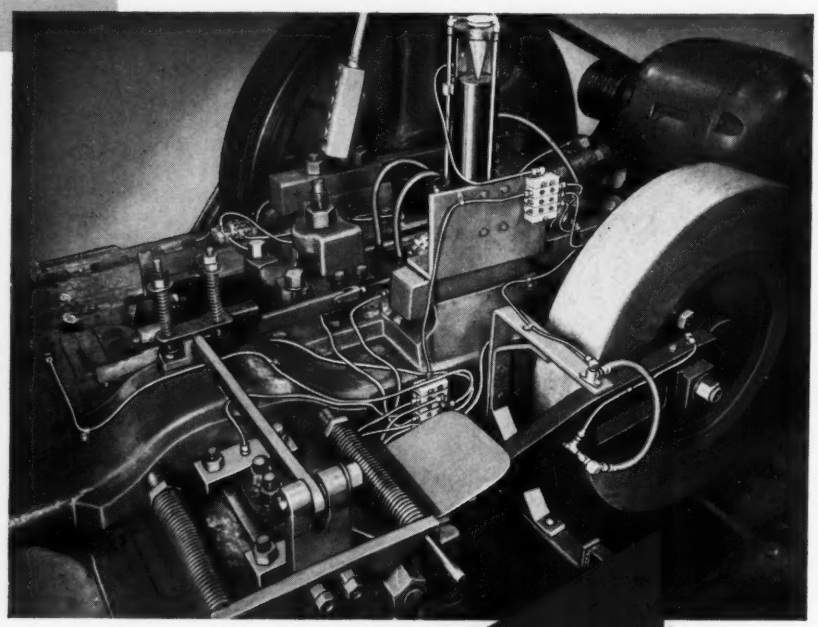

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Lubrication Engineering

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A·S·L·E

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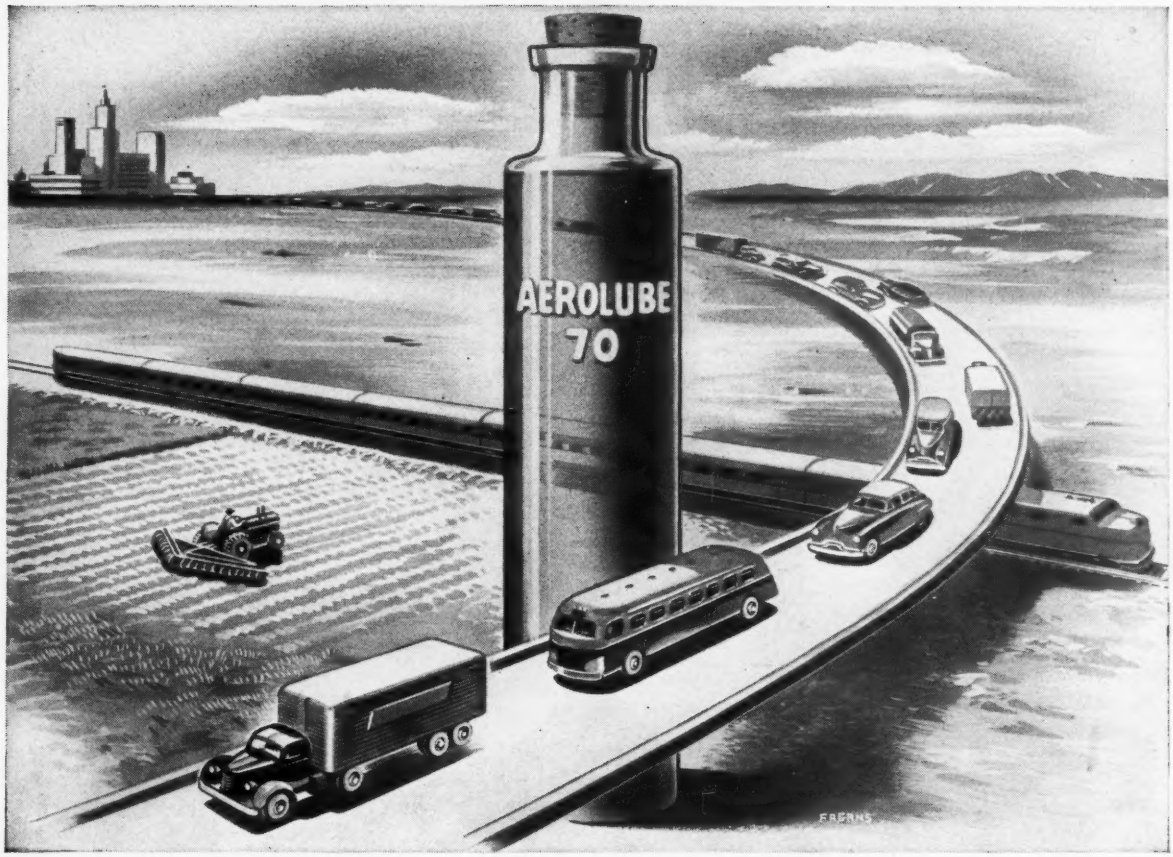

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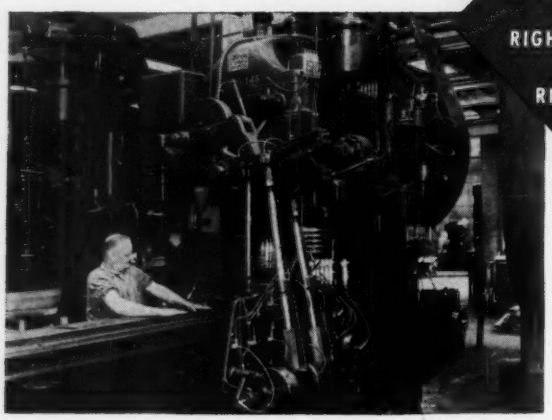

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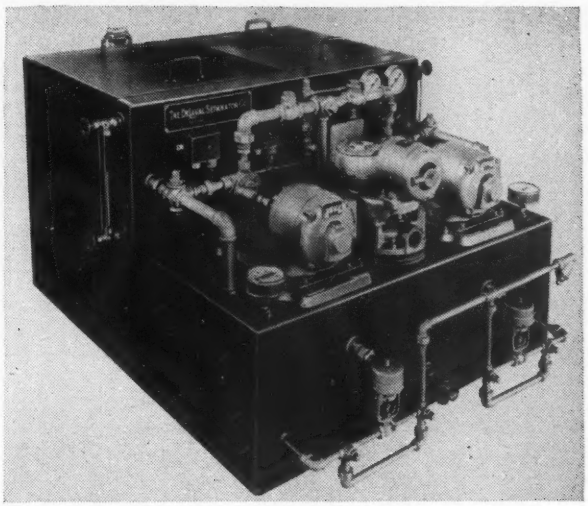

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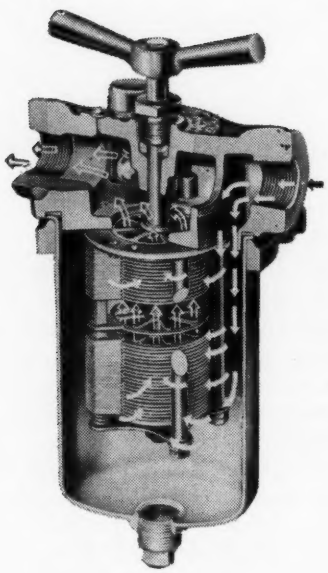

Single compact unit—no larger than usual partial-flow type—handles full flow of practically any fluid. Filtering element is permanent, all-metal, disc-type cartridge.

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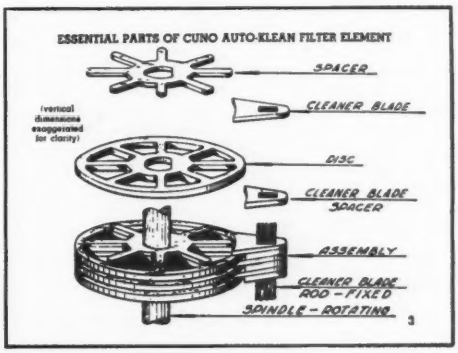

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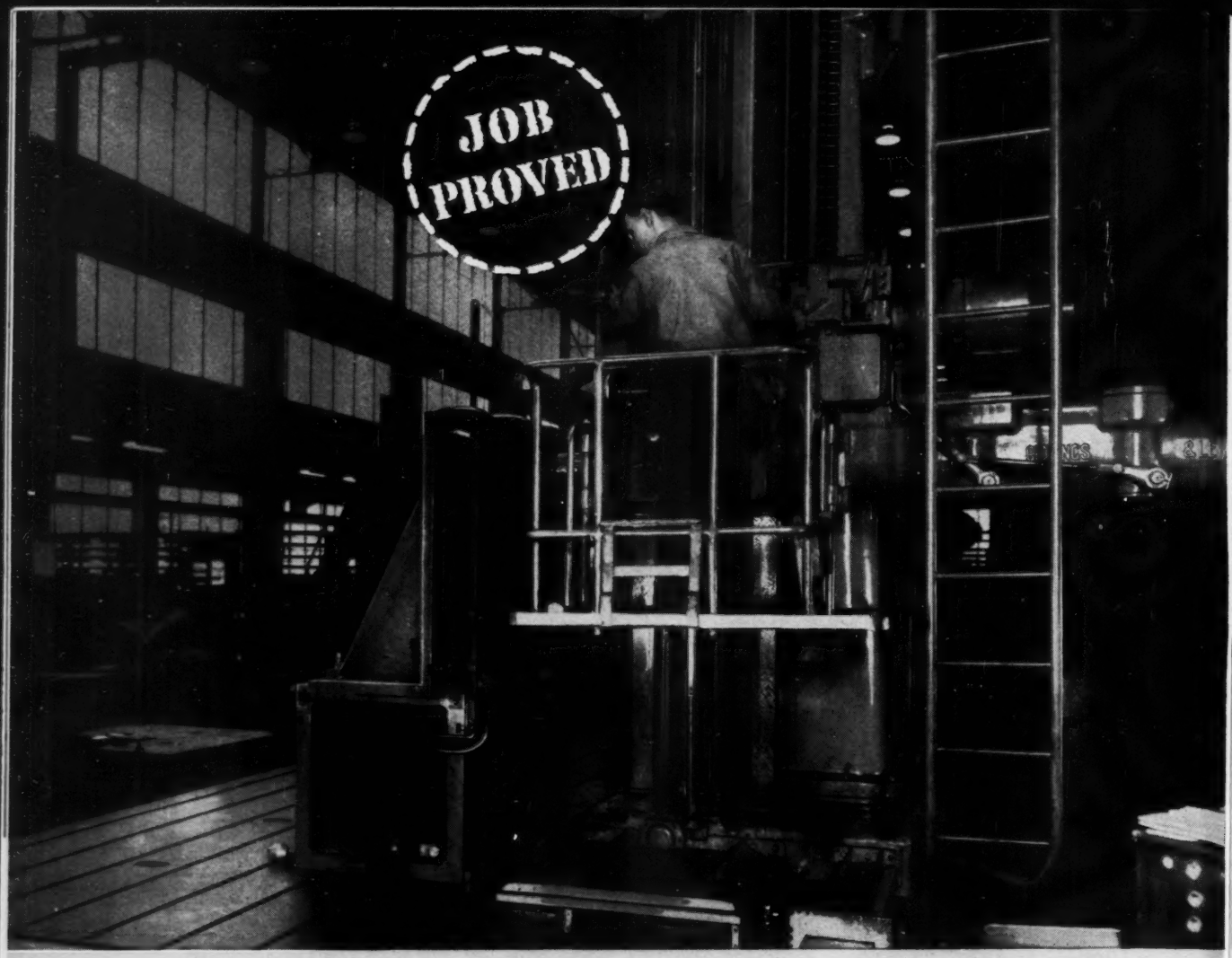

ENDS CHATTER, JUMP, DOWNTIME

Sunoco Way Lubricant Gives Superb Results After Other Products Are Tried and Rejected

A machine tool manufacturer was having trouble milling and boring steel plates which weighed up to 30 tons. Jump and chatter marred the finish and impaired the accuracy of the work. Every six months the department had to shut down for refinishing of the ways. All this added up to loss of production and costly maintenance.

Oil after oil was tried as a way lubricant, but the trouble continued. Then, acting on the recommendation of a Sun Engineer who had been called in, the company decided to try Sunoco Way Lubricant. In more than a year of use, this product has prevented all chatter and jump. Ways are in excellent condition and the quality of the work meets the manufacturer's high standards. A substantial sum has been saved through the elimination of downtime for way refinishing.

Sunoco Way Lubricant has exceptional metal-wetting and adhesive properties, as well as ample lubricity and extreme-pressure characteristics. Sunoco Way Lubricant resists corrosion. In shop after shop, it is helping to improve finish, increase production, and reduce costs. For complete information about Sun "Job Proved" Products refined for the metalworking industry, call or write the nearest Sun Office. The services of a Sun Engineer are always available to you without obligation.

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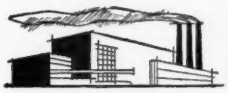

He's the meanest kind of thief—sly, slick and vicious as they come—stealing profitable production from machines in plants all over the country. But he can be stopped!

Perhaps you haven't even suspected that he works in your plant. Yet, that may be the *very reason* that you should *make sure* that he isn't!

LOOK BEHIND THESE MASKS

Are you still scooping grease from drums to guns by the old-fashioned "hand and paddle" method—wasting time, wasting grease, risking grime and dirt? There's a profit thief at work!

You can save 3¾ man hours for every 100 lbs. of grease.

Are you lubricating machines by hand—the old time-wasting, risky, hit-or-miss way? Watch for *big* losses here!

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Have you ever counted the losses on "downtime"—production interruptions for lubrication, part failures or replacements due to faulty lubrication? Unmask this thief!

You can lubricate hundreds of bearings from one central point, with the correct lubricant, in the right amount—while machines continue to produce.

Now are you suspicious that a profit thief is working in your plant?

Remember, his disguise is "old time," hit-or-miss, costly lubrication methods that add to the growing costs of production.

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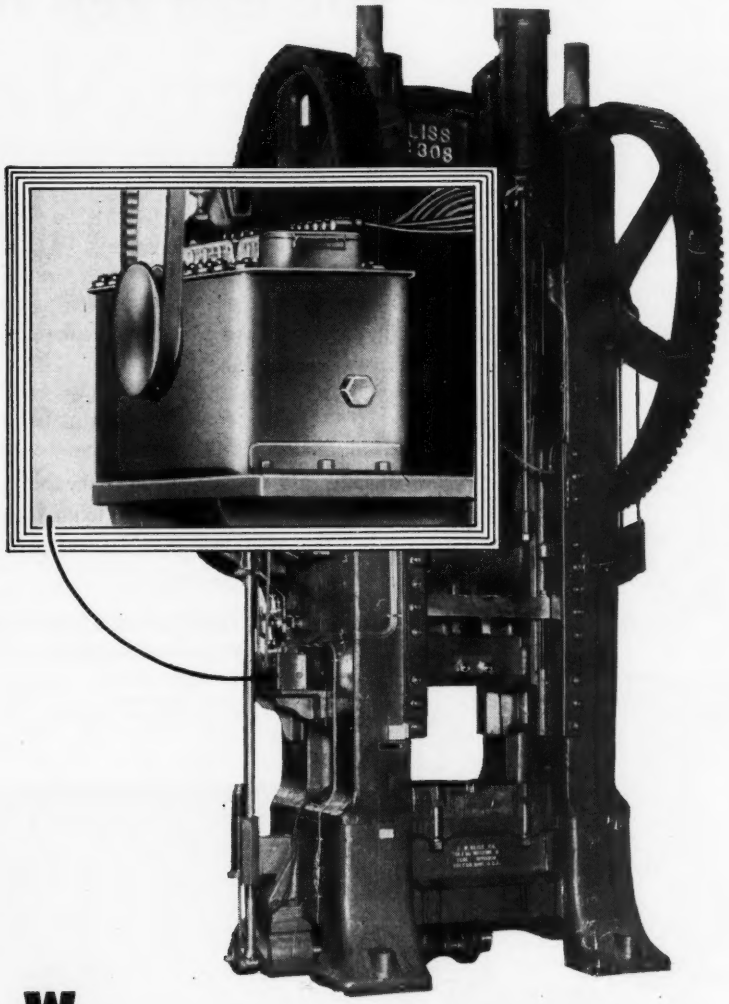

When the 20 hard-to-get-at bearings on this Bliss Crank Press were serviced by an oil can, it took 90 minutes a day at a labor cost of $2.10. In 10 years, with its modest initial cost taken out, this Manzel Model 25 Lubricator saves $5074.40 in labor alone. The oil saved by this non-wasting, never-forgetting guardian runs up to 60%...another *must* for Manzel Force-Feed Lubricators.

Manufacturers and users of pumps, engines and numerous types of heavy machinery have found Manzel Force-Feed Lubricators pay for themselves many times over. You probably have a lubrication job that can be handled much better and far more economically by Manzel equipment. Write us today for helpful information.

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Silicone News

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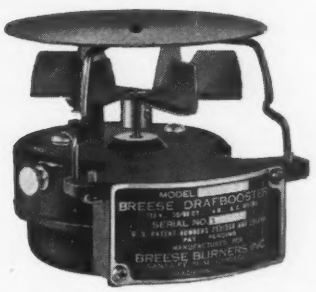

Photo courtesy Breese Burners, Inc.

Bearings of the BREESE DRAFBOOSTER oil furnace fan motor are lubricated with DC 200 Silicone Fluid to give years of satisfactory maintenance-free service.

Breese Burners, Inc., Santa Fe, New Mexico, manufactures the Drafbooster, one of the best fan motors available for oil-fired furnaces. Years of research and testing are incorporated into the design of this efficient unit and only high grade materials are used in its construction. Breese engineers learned, however, that in spite of well written instructions, few customers bothered to change oil each year. After two or three years' use, the oil was so badly oxidized and carbonized that the entire unit had to be replaced.

In 1944 Breese engineers obtained a sample of low viscosity DC 200 Silicone Fluid and subjected it to accelerated breakdown tests similar to those set up for testing the mineral oils Breese had been using.

M. D. Huston, Mgr. Sales Engineering for Breese, had this to say about the breakdown testing: "To our great surprise, the silicone fluid worked perfectly and showed no signs of oxidation or other deterioration. We immediately put Drafboosters containing DC 200 on lifetime field tests under the most severe operating conditions. A recent check of one unit which has been working since 1944 revealed the DC 200 Silicone Fluid is still just as good a lubricant as when it was put in. In August, 1948, we standardized on DC 200 Fluid for all our Drafboosters."

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THE EDITOR'S PAGE

In chatting with our national secretary, Bill Leonard, I was interested to hear him express some views on analogies. So much so, that I asked him to jot down some of his thoughts for this page. I think you will enjoy reading them.

"American civilization, industry and government have all changed greatly since the early days of our forefathers—in some ways for the better, in other ways for the worse.

In early days, the practice of lubrication was looked upon as a disagreeable necessity to keep the slow wheels of the machinery of that day turning. Outside water wheels were lubricated with chunks of bacon fat with the rind left on. The ways in ship-building yards were lubricated with over-ripe bananas to enable the ship to slip from its cradle over the skids into the water with a minimum of friction. Crude methods such as these were sufficient for the needs of the time.

In looking at our present-day facilities for the manufacture of lubricants and methods for their application, we find that many millions of dollars have been poured into research laboratories and into the maintenance of many scientists so that now there are few problems which cannot be overcome instantly with up-to-date petroleum and synthetic products. Each of these products can be applied with practical and workable apparatus designed for a special purpose, with the result that the complicated machinery of today operates with a minimum of friction, insuring few interruptions in production.

We have progressed mechanically from necessity. Although we have the same need in our governmental machinery, we are not meeting that need.

When our government came into existence in the early days of the Philadelphia assembly, our entire country was composed of only 13 states. Communication was difficult and the means of continuing organization without breakdown would be considered extremely crude, by present day standards. Accompanying our growth, we have become a tremendously complex organization, and government, which unfortunately is our leading industry, is over-burdened by intricate bureaucratic organization beyond the understanding of any individual.

We have eliminated, from our early days, the creaking from inefficient lubrication, but we are now faced with the problem of National and International friction, and we hear again the squeaking, rattling, the dry sound of our heated political bearings. It would seem to the casual observer that our present government is seeking to perpetuate itself in an ever-increasing maze of complexity, and he might wonder if it were not seeking to work out the fallacious theory of perpetual motion in a frictionless paradise for governing bodies.

Industry has had the benefit of lubrication engineers. Our government must have the benefit of human and economic engineers in order to perpetuate our free system of enterprise with smooth running governing machinery. We need men comparable to those of 1776; men with the vision to serve without self-interest or allegiance to party, who will direct their efforts to the orderly processes of government for, by and of the people, guiding us back to the orderly channel of self-accomplishment.

Where can we find the human engineers to solve the problems of government?"

The recent appearance of what seems to be a really successful gas turbine locomotive has caused a new stir of interest in this novel development. This locomotive, jointly built by the American Locomotive Company and General Electric, is a 4500 HP unit. It successfully completed road tests with some of the Eastern railroads and is now undergoing a severe test program with a Western railroad.

This particular unit has a straight through arrangement of compressor combustion chambers and turbine. The turbine not only drives the compressor, but with its surplus power turns the generator which supplies the power to eight traction motors. It is interesting to note that the very successful mechanical electrical system used with diesel locomotives has been adapted for this gas turbine locomotive as well. The turbine and compressor run at 6700 rpm at normal load condition with 1400 F inlet temperature to the turbine. The locomotive is started on diesel fuel, but after the warm-up period operates with No. 6 grade fuel oil. Thermal efficiencies in the neighborhood of 18% have been reached with this unit. Such values are, of course, lower than those obtained with diesel locomotives, but the ability of the gas turbine to use the lower grade residual fuels, and at some future date even solid fuels in powdered form, more than offsets this lower efficiency.

We now appear to be on the threshold of seeing a new type of unit take its place in the power world. The old iron horse of our youth is being set back another step!

The Government is now endeavoring to complete a new roster of qualified engineers and scientists with an extensive arrangement of cross-referencing so that our wealth of manpower can best be allocated in case of an emergency. Many of our engineers have already received application forms to be filed for listing in this connection, and those who have not will probably receive them in a short time. The Office of Naval Research has been assigned the job of coordinating the information being compiled. This new roster is similar to one which was prepared during the War period.

DYNAMICALLY LOADED BEARINGS*

by Milton C. Shaw
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Professor Shaw received his undergraduate training in mechanical engineering at Drexel Institute of Technology. During the next four years he was a Cincinnati Milling Machine Company Fellow at the University of Cincinnati where basic work in chemistry, physics and mathematics was taken leading to the Doctor of Science degree. Professor Shaw then worked in the research laboratories of the Cincinnati Milling Machine Company. During the war years from 1942 to 1946, he was associated with the National Advisory Committee for Aeronautics, first at Langley Field, Virginia and later at the Flight Propulsion Research Laboratory in Cleveland. In 1946, Professor Shaw left his NACA position as Chief of the Materials Branch to assume his present position as Associate Professor of Mechanical Engineering at MIT. Professor Shaw is a member of the American Society of Mechanical Engineers, American Chemical Society and the American Physical Society.

ABSTRACT

In the past the hydrodynamic theory of lubrication of the conventional journal bearing, in which the shaft rotates at constant angular velocity while the bearing is subjected to a load that is invariant in magnitude and direction, has received considerable attention. However, a good many bearings that are encountered in practice do not operate under these relatively simple conditions; many practical bearings are subjected to rotating and fluctuating loads and frequently both bearing surfaces are in motion. This paper discusses such unsteady state problems with a minimum of mathematical profundity. The method adopted consists of reducing each system considered to an equivalent conventional journal bearing for which solutions are readily available.

A hydraulic analogy is presented for the dynamically loaded bearing which is useful in qualitative analysis. The operating conditions for a piston-pin bearing of a reciprocating engine are considered in detail in order to illustrate the principles discussed and their method of application. While the bearing considered is found to operate under a maximum unit load of 9400 psi, the equivalent conventional journal bearing is subjected to a unit load of but 1520 psi, which is a reasonable value for a well designed conventional bearing. The very high load capacity of the piston pin bearing is thus accounted for by its dynamic conditions of operation.

Introduction

In all current texts concerning hydrodynamic lubrication theory the full journal bearing shown in Fig. 1 is considered in detail. The shaft of this unit rotates at constant angular velocity while the bearing supports a load that is constant in magnitude and direction. By expressing the law of conservation of matter, Newton's second law ($F_x = ma_x$) and the defining equation for viscosity in mathematical form, the following expression is obtained for bearings of the type shown in Fig. 1.

$$\frac{\partial}{\partial x}\left(\frac{h^3}{6\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{6\mu}\frac{\partial p}{\partial z}\right)=U_1\frac{\partial h}{\partial x} \qquad (1)$$

where

h is the oil film thickness at any point
μ is the viscosity of the lubricant
p is the pressure in the oil film at any point
U_1 is the velocity of the journal surface
z and z are rectangular coordinates in the circumferential and axial directions respectively.

This rather formidable expression is known as Reynolds' Differential Equation and has caused a good many engineers to shudder upon encountering it for the first time. This equation is not difficult to solve for the hypothetical case of a bearing of infinite length which has no pressure gradient along its length. However, the solution for a bearing of finite length is a very difficult matter and many able mathematicians have labored long to obtain an approximate solution for a single set of operating conditions. The practical way of applying the hydrodynamic equations to bearing design is to first use the no-end flow theory and then apply a suitable correction factor to take care of the reduction in load capacity due to the drop in pressure along the bearing as a result of end flow.

When Reynolds' equation (1) is solved for a bearing having no end flow and constant viscosity, the load capacity W is given by

$$W=\mu U_1 L\left[\frac{r}{c}\right]^2\left[\frac{12\pi n}{(2+n^2)(1-n^2)^{1/2}}\right] \qquad (2)$$

where

L is the axial length of the bearing
r is the radius of the bearing
n is the maximum allowable attitude of the bearing

$$n=\frac{\text{Maximum allowable eccentricity}}{\text{radial clearance}}=\frac{e_{max}}{c}$$

* Paper presented at 4th Annual Convention of A.S.L.E., New York City, April 12, 1949.

Now these results are well known and are thoroughly discussed in many texts on Machine Design, Fluid Mechanics and Lubrication. Equation (2) is in good qualitative agreement with experimental results and may be brought into good quantitative agreement by use of an end-leakage correction factor as already mentioned.

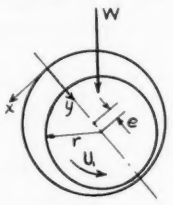

Fig. 1

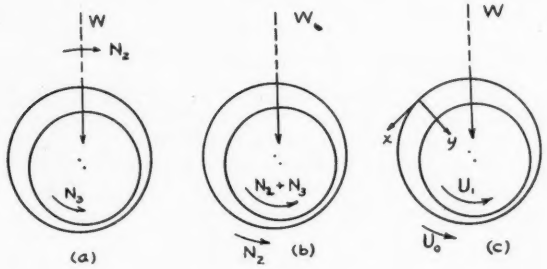

Fig. 2

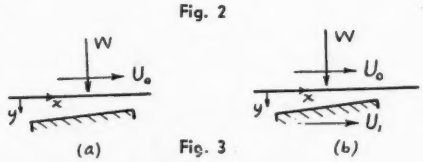

Fig. 3

Dynamically Loaded Bearings

Unfortunately many of the bearings encountered by the design engineer are not as simple as that shown in Fig. 1. For example most of the bearings found in an internal combustion engine rotate at speeds that vary periodically in magnitude and sometimes even in direction, and at the same time are subjected to loads that vary in both magnitude and direction. Such a general case is shown diagramatically in Fig. 2a. In practice it is more convenient to consider the kinematically equivalent diagram shown in Fig. 2b which is drawn relative to a fixed point on the load vector instead of a fixed point in space. The theory found in most lubrication texts does not apply to this more general problem.

An extended form of Reynolds' equation may be derived for the more general problem of Fig. 2c following a sequence of steps similar to those used to obtain equation (1). The resulting expression is

$$\frac{\partial}{\partial x}\left(\frac{h^3}{6\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{6\mu}\frac{\partial p}{\partial z}\right)=(U_0-U_1)\frac{\partial h}{\partial x}+h\frac{\partial(U_0+U_1)}{\partial x}+2V_1 \quad (3)$$

where

U_0 and U_1 are surface velocities as shown in Fig. 2c.

V_1 is the normal velocity of the journal surface relative to the bearing surface at any point.

In applying this equation, it is convenient first to evaluate the right hand side from a consideration of the geometry of the bearing. For example, consider the plane slider bearing of the Kingsbury type shown in Fig. 3a. This bearing is subjected to a load of constant magnitude. Equations (1) and (3) are both applicable to this bearing and in this case

$$U_0 = U_0$$
$$U_1 = O$$
$$V_1 = O$$

and equation (3) reduces to equation (1). Next consider the slider bearing shown in Fig. 3b in which both surfaces are in motion. In this case

$$U_0 = U_0$$
$$U_1 = U_1$$
$$V_1 = O$$

and the right side of equation (3) becomes

$$(U_0 - U_1)\frac{\partial h}{\partial x}$$

If the speeds of the two surfaces are the same this quantity is zero and the load capacity would be the same as though the bearing was not in motion. The quantity $(U_0 - U_1)$ is the relative velocity of the two surfaces and hence it is evident that the load capacity of a slider bearing with both surfaces in motion is *proportional to the relative velocity of the surfaces* just as the load capacity of the ordinary slider with one surface in motion is proportional to the velocity of that surface.

By direct analogy the load capacity of a journal bearing might be expected to vary directly with the relative velocity between the bearing and journal surfaces. The bearing of Fig. 2c would thus be expected to be zero if U_1 and U_0 were equal. Such is not the case however and this difference in behavior between slider and journal bearings is worth noting. For the bearing of Fig. 2c

$$U_0 = U_0$$
$$U_1 = U_1$$

$$V_1 \cong U_1\frac{\partial h}{\partial x}$$

and the right side of equation (3) makes U_0 and U_1 additive.

Comparing this quantity with the right side of equation (1), it is evident that the load capacity of a journal bearing with both surfaces in motion *is proportional to the sum of the two surface velocities.* Stone and Underwood (1) have experimentally verified this result.

Hydraulic Analogy

A hydraulic analogy has been devised (2) to help keep the relative load capacities of dynamically loaded bearings straight. A dynamically loaded bearing behaves as though it were a pump, a valve, and a piston all at the same time. Physically these elements are superimposed in the actual bearing to such an extent that they bear little resemblance to conventional pumps, valves and pistons. However, the dynamically loaded bearing shown in Fig. 4a may be likened to the hydraulic system shown in Fig. 4b. This system consists of two gear pumps G_0 and G_1, which may be operated at any speed in either direction of rotation; a piston P; a pressure gage B; and a spring supported valve V. The pressure developed at the point B is considered analogous to the load supported by an equivalent journal bearing. The rotating shaft of an ordinary unidirectionally loaded journal bearing acts as a pump

in that it causes oil to flow past the portion of the bearing under the load. Gear pump G_1 therefore represents the pumping action of a rotating shaft in the analogy. Likewise, a rotating bearing can transport oil past the portion of the journal under the load; this is similarly represented in the analogy by gear pump G_0.

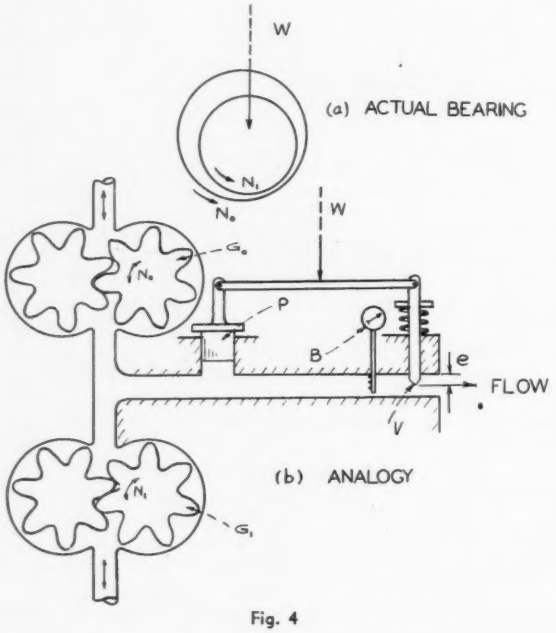

Fig. 4

A journal is known to move eccentrically to the bearing as load is applied, causing the bearing surfaces to approach each other in the region under the load. This action restricts the flow of oil and hence causes sufficient pressure to be developed to support the load. The spring-supported valve V is introduced in the analogy to restrict flow with an accompanying rise in pressure at B. It is interesting to note that the valve spring has a nonlinear-deflection characteristic, since the eccentricity of a journal does not vary linearly with load.

The piston P is introduced to simulate the influence of a fluctuation load upon the load capacity of a journal bearing. It is evident that a sudden increase in load W produces a corresponding increase in the pressure at B and hence an increase in the load capacity of the equivalent journal bearing.

The analogy will be considered first from the standpoint of a bearing operating under a load of constant magnitude. In such a case the piston has no function and will therefore be ignored in the discussion immediately following. The ordinary journal bearing is represented by a counterclockwise rotation of G_1, with G_0 stationary. The bearing is incapable of supporting load as long as the journal runs concentrically with the bearing, for in this case valve V is wide open and a negligible pressure is registered at B. In order to support load the journal must move to an eccentric position corresponding to a nearly closed position of valve V. An increase in the speed of the pump (journal) or viscosity of the lubricant will cause an increase in the pressure at B for a given setting of valve V (eccentricity). Hence, the valve need not be closed so far to develop a given pressure (i.e. support a given load) when an oil of increased viscosity or an increased pump (journal) speed is used. Simultaneous rotation of both journal and bearing in the same direction (G_1 counterclockwise and G_0 clockwise) and at the same speed will cause twice as much flow in the analogy, and hence twice as much pressure will develop at B for a given position of valve V (eccentricity). If the bearing is rotated at the same speed as the shaft but in the opposite direction, the net flow will be zero and the analogy predicts zero load capacity.

Many bearings are subjected to a periodically fluctuating load. For such cases the piston P must be employed. It is evident that the pressure at B (and hence the load capacity of the bearing) will be increased by the effect of the piston when the load is increasing and conversely. It is also evident that a periodically fluctuating load of high frequency will have a greater influence upon the pressure at B than a fluctuating load of low frequency.

Fluctuating Load — No Rotation

Equation (3) cannot be solved in closed form when even the simplest type of fluctuating load is imposed on a full journal bearing of infinite length. In fact it is about as difficult to obtain an approximate solution in this instance as in the case of the bearing of finite length operating under a load of constant magnitude. Both of these analyses are impractical from a design engineering point of view. However, the case of a bearing that is subjected to a fluctuating load with neither surface in rotation and without end flow can be readily solved. Equation (3) then becomes

$$\frac{d}{dx}\left(\frac{h^3}{6\mu}\frac{dp}{dx}\right) = 2V_1 = 2\frac{dh}{dt} \qquad (4)$$

By following a series of steps similar to those used to go from equations (1) to (2), equation (4) is found to give (3)

$$W = 12\pi\mu r L\left(\frac{r}{c}\right)^2 \frac{1}{(1-n^2)^{3/2}}\frac{dn}{dt} \qquad (5)$$

If the load fluctuates sinusoidally as shown in Fig. 5, then

$$W = W_0 \sin \omega t \qquad (6)$$

where W_0 is the amplitude of the load ω is a uniform angular velocity. When equations (5) and (6) are combined and integrated the following result is obtained for the load capacity of the bearing

$$W_0 = \mu r \omega L\left(\frac{r}{c}\right)^2 \frac{12\pi n_1}{(1-n_1^2)^{1/2}} \qquad (7)$$

where n_1 is the maximum allowable attitude.

The Piston-Pin Bearing

The piston-pin bearing of a reciprocating engine (the bearing at A in Fig. 6) will now be considered to illustrate the manner in which the foregoing equations might be used to interpret the complex loading cycle of this bearing. The load on the piston-pin bearing comes from two sources (1) the gas pressure acting on the top of the piston and (2) the inertia force due to the reciprocating motion of the upper end of the connecting rod. These two force vectors may be readily computed for any position of the crank by applying the laws of elementary mechanics (2) and when combined vectorially yield the resultant force on the piston pin. It is customary to plot the results of such computa-

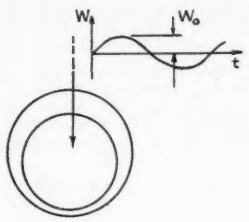

Fig. 5

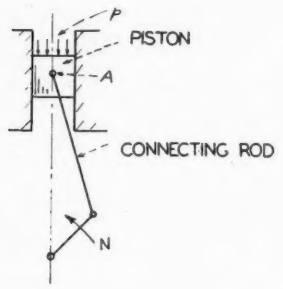

Fig. 6

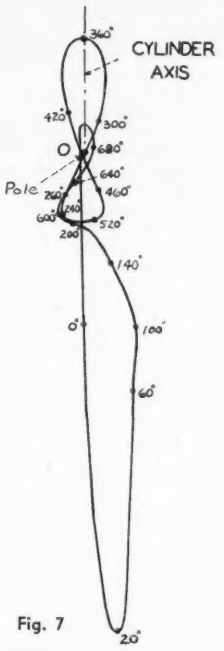

Fig. 7

tions for different crank angle values in the form of a polar diagram as shown in Fig. 7. A line drawn from the pole O to any point on the curve represents the resultant bearing load vector in magnitude and direction relative to the cylinder axis, zero degrees being the crank angle value corresponding to the top center position during the power stroke. The specific diagram shown in Fig. 7 is for the piston pin of an in-line aircraft engine operating under the following conditions:

Bore, in.	5.50
Stroke, In.	6.00
Indicated mean effective pressure, psi	300
Engine speed, rpm	3000
Mechanical efficiency	0.77
Compression ratio	6.65
Connecting rod length, in.	10.00
Reciprocating weight per cylinder, lb.	6.72

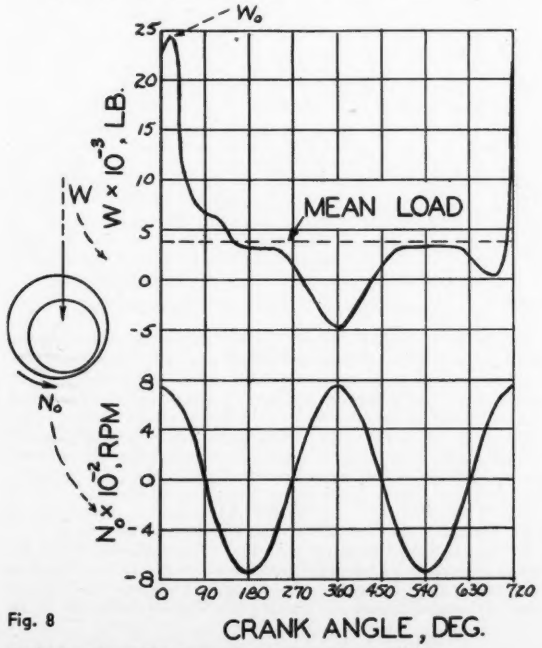

Fig. 8

From Fig. 7 it is evident that the direction of the bearing load vector is essentially along the cylinder axis for all values of crank angle and hence the slight rocking motion of the load vector may be ignored. Thus, the load acting on the piston pin bearing of this engine is found to vary periodically as shown in the upper curve of Fig. 8. The piston pin is affixed to the piston bosses and the upper end of the connecting rod executes an angular reciprocating motion as shown in the lower part of Fig. 8. This figure is representative of the conditions under which most reciprocating engine bearings operate, and it is obvious that such a situation is far different from that of the bearing shown in Fig. 1. The maximum load to which this bearing is subjected is 23,500 pounds. The projected area of the bearings operate, and it is obvious that such a situation unit bearing load is 9400 pounds per square inch. This is a very high unit load and a bearing of the type shown in Fig. 1 probably could not long withstand it.

Fogg and Jakeman (4) have experimentally studied the lubrication characteristics of a bearing subjected to a load of constant magnitude and direction and a sinusoidally oscillating relative circumferential motion between the bearing surfaces. Lubrication was found to become more hydrodynamic as the frequency of the oscillation was increased, but under no conditions of operation up to a frequency of 1000 cycles per minute was complete fluid lubrication reached. We should therefore look elsewhere for the explanation of the high capacity of the piston-pin bearing.

Consider the non-rotating bearing shown in Fig 9 which is subjected to a sinusoidally fluctuating load. The frequency of the load fluctuation ($\omega = \frac{2\pi}{3}$ radians per second, is seen to correspond closely to that for the maximum load peak of Fig. 8. The maximum attitude corresponding to the condition of loading shown in Fig. 9 is found to be 0.26 from equation (7) for the following bearing operating conditions

$$\frac{r}{c} = 1000$$

Viscosity, lb. sec. sq. in.	=	2×10^{-6}
Maximum load, W_0 lb.	=	23,500
Bearing area, sq. in.	=	2.50

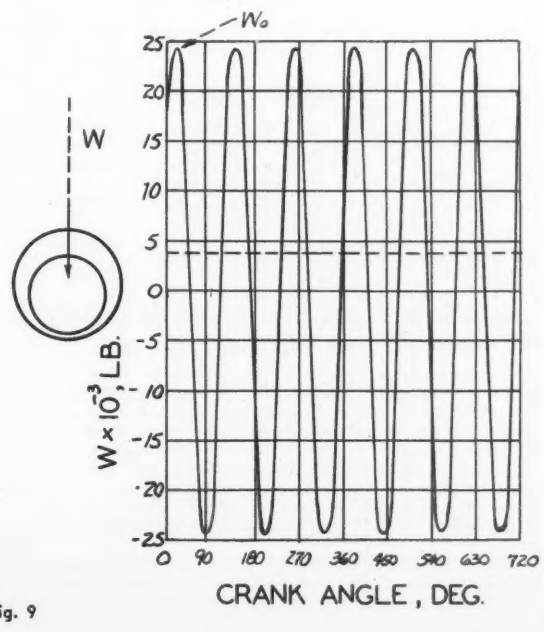

Fig. 9

The constant load required to produce this attitude in a conventional bearing (Fig. 1) of the same proportions as the piston-pin bearing when operating at a speed of 3000 rpm is seen to be but 3800 pounds (or 1520 psi) from equation (2). It is thus evident that a bearing subjected to a sinusoidally fluctuating load of high frequency may have several times (in this case 6.2 times) the load capacity of the equivalent conventional type-bearing rotating at one third the frequency of the loading cycle. A unit load of 1520 psi is generally not considered excessive for a conventional journal bearing, and hence it becomes evident that in the case considered here a peak unit load of 9400 lb/in^2 is not apt to cause trouble.

While the oscillatory motion of the piston pin is probably unable to achieve complete hydrodynamic lubrication alone, it undoubtedly contributes to the overall performance of the piston pin bearing.

Usually the assumption of zero flow from a bearing gives rise to major discrepancies in lubrication calculations and the results are only of qualitative value. In the procedure adopted here complex problems are transformed into their conventional journal bearing counterparts. Since no-end-flow theory is used throughout the discussion, the discrepancies which usually arise due to the no-end-flow assumption are largely neutralized when the comparison is made. While the load capacity of the actual bearing and the equivalent bearing will both be too high as determined by the simplified theory, their ratio, which is the only quantity on which conclusions are based, will be in much better relation to truth than the values constituting the numerator and denominator of the ratio. This is one of the few ways of effectively using uncorrected no-end-flow lubrication theory.

It is evident from the foregoing example that the bearing designer must use care in interpreting the values obtained from a polar load diagram. While hydrodynamic bearing theory is not capable of yielding a precise solution to the complex situations that are frequently met in practice, it can generally be employed qualitatively as a powerful adjunct to extensive practical experience and keen engineering judgment.

The writer wishes to acknowledge valued discussion with Mr. E. Fred Macks of the National Advisory Comittee for Aeronautics in the development of some of the ideas set forth in this paper.

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Discussion by R. J. S. Pigott and J. M. Cunningham

Gulf Research and Development Co.

The author is correct in his statements that imply the major attention in lubrication theory has been directed on the hydrodynamic theory as applied to a journal bearing under a load of constant magnitude and direction. Considering the millions of journal bearings operating in internal combustion engines and other reciprocating machines on which the conditions of constant load magnitude and direction do not obtain, it would appear that any method of analyzing lubrication in a dynamically loaded bearing should command considerable attention.

In a dynamically loaded bearing, the journal center has a cyclic motion of its own apart from the journal rotation, so that the load supporting portion of the lubricating film with its maximum and minimum pressure points is shifting rapidly in space phase around the bearing. It is not easy to visualize nor to estimate the end results from this periodic disturbance recurrence in the rupture and reforming of the load supporting film at successive points. Under specific conditions certain of these phenomena will admit to mathematical treatment but mathematical analysis alone is insufficient. There is a need for other devices of the order of an analogy in order to obtain a satisfactory understanding of the operation.

This manuscript describes a form of a hydraulic analogy that is unique in its representation of certain phases of a dynamically loaded bearing. It appears that within its limitations the analogy will successfully illustrate certain lubrication phenomena occurring in a journal bearing.

The analogy as described unfortunately has limitations in that it will yield only relative values of load capacity. Furthermore, it is incapable of representing the other important characteristics and conditions of friction, temperature, oil flow and attitude in a dynamically loaded bearing. In addition, the relative load capacity has certain restrictions. For example, the analogy is limited in representation of relative load capacity of a fluctuating load to a load only in a fixed direction, although the fluctuating load may be in combination with any positive or negative velocities at the bearing and journal surfaces. The analogy will illustrate also the relative load capacity of a rotating load provided that the load is constant in magnitude and rotational velocity. Here also the rotating load may be in combination with any fixed velocities of the bearing and journal surfaces. The analogy is incapable of indicating even qualitatively, the power loss from viscous shear in the lubricating film, or of indicating the relative bearing temperature and relative flow rate of the lubricant through a bearing. Also, it appears that due to inertia effect in the analogy gear pumps and fluid, the analogy cannot represent relative overall effects on load capacity from rapid reversals of rotation of the bearing and journal surfaces such as occur at the crankpin end of a blade rod. However, here the limitation of inertia effects possibly may be overcome by substitution of cam driven piston pumps or the gear pumps.

The action of the two gear pumps should illustrate nicely the relative lubrication effect with reference to load capacity that occurs in many gear train bearings wherein both the journal and bearing rotate at separate velocities in either the same or opposite directions.

With further reference to the relative load capacity of a constant rotating load in combination with fixed velocities of the bearing or journal surfaces, the analogy will illustrate this type of load provided the rotation of the two gear pumps is selected to represent the dynamical equivalents of the bearing and journal surface velocities. The relative load capacity being indicated proportional to the sum of the equivalent surface velocities of the bearing and journal, there are some interesting results obtained with this type of loading. Using the notation of positive values of N_1, N_2, and N_3 to represent clockwise velocities of the bearing surface, rotating load, and journal surface respectively, and the negative values to represent their counterclockwise velocities. Then with positive values of N_1, N_2 and N_3 and referring the velocities to the load vector, or line of centers of the bearing and journal, the equivalent surface velocity of the bearing will be $(N_1 - N_2)$, the equivalent surface velocity of the journal will be $(N_3 - N_2)$, and the relative load capacity given by their sums will be proportional to $(N_1 - 2N_2 + N_3)$. From this result it is evident that there are many combinations of velocity direction and velocity magnitude which will make this sum zero, and hence the load capacity zero is theory. For example, with a stationary bearing or zero velocity for N_1 and with the load rotating at half the velocity and in the same direction as the journal, then the sum of the equivalent surface velocities of the bearing and journal and hence the relative load capacity become proportional to $N_3 - 2N_2 = N_3 - \frac{2N_3}{2}$ and is equal to zero. Conversely, other combinations of velocity direction and velocity magnitude will yield an increase in the relative load capacity.

With regard to fluctuating loads in a fixed direction, it appears that there should be two opposed pistons interconnected through linkage to illustrate relative load capacity.

There is noticed a slight error, probably typographical, in the right hand side of equation 7 which may have escaped the author's attention. The terms under the radical in the denominator are reversed and should read $\sqrt{1 - n^2}$.

In conclusion, the author is to be complimented on his hydraulic analogy which is a contribution to the comprehending of the difference in relative load capacity between a bearing loaded dynamically and one having a constant stationary load.

Discussion by Roy E. Edwards

Gardner-Denver Co., Quincy, Illinois

Before beginning the discussion I would like to state that I am one of those engineers easily frightened by a differential equation, consequently, for the present I cannot support the following comments by a strict mathematical analysis. I might add that most of my experience with problems discussed in Professor Shaw's paper has been with slow speed (50 to 150 RPM) high pressure, single acting and double acting, oil field pumps with piston or plunger loads from 50,000 to 200,000 pounds.

Under these conditions, we believe the wrist pin bearing acts similar to a double acting pump discharging through two parallel lines. In this case the hydrodynamic effect due to rotation can be neglected, we believe, especially if width to diameter ratio is less than one.

If we neglect the rotational component, which will probably be super-imposed on that of the squeeze component, the "pressure field" would probably be reduced to a condition as indicated in Fig. A, and which would be symmetrical because the distortion effect of the rotational component would be eliminated. Hence we believe that the total discharge capacity Q_t or $Q_z + Q_x$ can be expressed as a function of the rate of load application, clearance and viscosity of the lubricant, this determining the shock absorbing effect. Referring to Fig. A, Pmax would be the maximum allowable pressure developed at the critical film thickness.

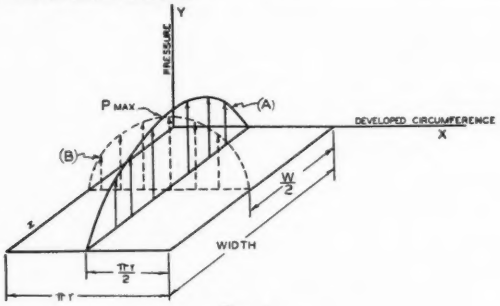

Fig. A

The time for the pin to travel from one side of the bearing to the condition of critical film thickness on the opposite side as compared to the time required for one cycle of angular motion of the rod is important. This condition should not be attained before the end of one angular cycle. If it is reached before this time the bearing might "burn" because of the intense pressure and relative motion of the surface projections of the mating parts.

For these reasons, for a wrist pin bearing, we favor the theory based on the velocity of approach of two plates with a viscous fluid between them. Such a theory has been advanced by Dr. Dudley Fuller of Columbia University. This theory by Dr. Fuller neglects side leakage, something we do not believe can be safely done. However, if this theory were modified to include the above factors, we believe it would be entirely satisfactory.

From Fig. B, this discussion, we can see that the variation of the area along the length of the bearing is affected a negligible amount by considering the path of the center-point a straight line, along the axis of the ellipse which we consider to be the true path. Note that as the diameter of the bearing pin approaches that of the bushing the variation in this area becomes less, consequently, we can probably assume the area of this "flow" path to be constant.

There will be a variable "flow" path that varies with the rate of application of the load. This latter path controls side leakage.

Borrowing the idea of a hydraulic analogy from Professor Shaw, we offer the sketch Fig. C, which we believe more accurately portrays the action of a wrist pin bearing. On the basis of the above assumption, Fig. B, the area A_x (lengthwise area) can be represented by a fixed orifice and the action of the side area A_z by the piston and valve linkage. Then P and P_1 represent the squeeze pressure and the rotational pressure respectively. From this we can see that it is entirely possible for the squeeze component to nullify the rotational component if of sufficient magnitude.

Fig. D, portrays the action of the crankpin bearing as we see it. From the sketch it would seem that the squeeze component tends to counteract the effect of the rotational component. Depending on the relative magnitude of the two, it might be conceivable that we could have zero film thickness.

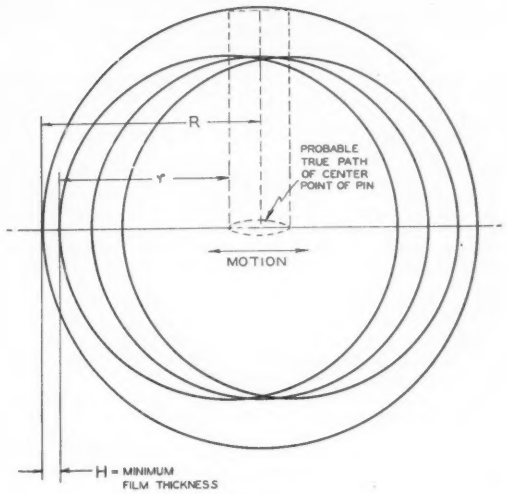

Fig. B

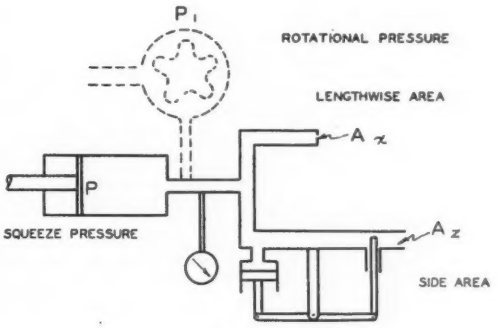

Fig. C

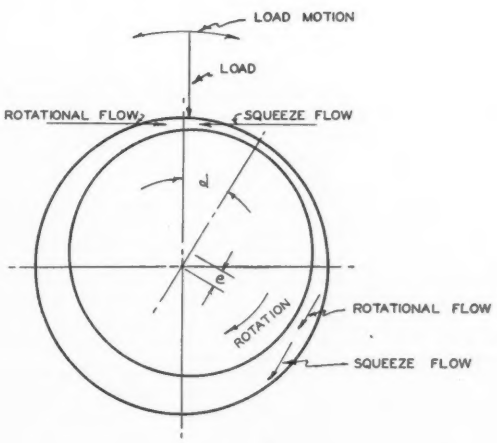

Fig. D

The effect of the squeeze component tends to increase the eccentricity and the angle a. This means that the maximum possible attitude as stated by Professor Shaw would have to include this condition, except for the time involving zero film thickness which would probably be of extremely short duration.

Discussion by Joseph B. Bidwell

Research Laboratories, General Motors Corporation

Mr. Shaw's paper should be extremely interesting to anyone concerned with bearing design in piston engines. The success or failure of bearings in such applications often depends much more on the character of the load diagram than upon the maximum loads.

The simplicity of wrist pin bearings for four cycle engines as compared with the same application in a two cycle engine where the load is not reversed in direction is an outstanding example of the importance of what Mr. Underwood has termed the "squeeze" film. That is the load carrying oil film produced under dynamic reversing load conditions with no rotation of journal or bearing.

The variation of relative velocity between journal and bearing as described by Mr. Shaw often accounts for the ability of main and rod bearings to carry peak dynamic loads many times that under which they could steadily operate.

Mr. Shaw's approximation of the maximum attitude of the wrist pin bearing considered in his paper should be modified somewhat. In his calculation, Mr. Shaw has used what he intends as an equivalent load diagram. This load is sinusoidal with a magnitude equal to the maximum load and an angular velocity three times shaft speed. The peak of this curve is similar in appearance to the maximum load peak of the actual load diagram.

This analysis neglects the effects of the somewhat smaller loads acting for some time after the maximum has occurred as well as the effect of load reversal. The importance of these factors will be evident from the following observations.

First, it should be noted, that the direction of motion of the journal is always in the direction of the applied force if inertia forces are neglected. This is a consequence of the equilibrium of forces acting on the journal. The assumption of a sinusoidal load diagram at three times shaft frequency would indicate the maximum attitude to be reached at about 50° crank angle at which time the direction of motion would reverse. The actual load diagram, however, shows that the load does not reverse until about 290° crank angle. During this difference of some 240°, an average load of about 3500 pounds is continuing to reduce the minimum film thickness beyond that calculated by Mr. Shaw.

The effect of this smaller load acting for such a long time interval may be best seen by rearranging equation 7 of Mr. Shaw's paper so that all of the terms involving eccentricity will be on one side.

$$\frac{n_1}{(1-n_1^2)^{1/2}} = \left[\frac{1}{12\pi\mu r h \left(\frac{r}{c}\right)^2}\right] \frac{W_o}{\omega}$$

The other side of the equation is then the product of a term composed of the characteristics of the oil and the bearing geometry and the ratio, $\frac{W_o}{\omega}$.

It is readily seen from this relation that a fraction of the maximum load can produce the same attitude if its frequency is lower or time of application greater. It is, therefore, not desirable to neglect the lower load applied up to 290° crank angle.

The importance of the magnitude and time during which the reversed force acts should also be emphasized.

The substitution of the completely reversed sine load for the actual bearing load diagram means that the maximum attitude will be the same for both loading directions. This will not be true in the case of the actual load diagram since the reversed force is low in magnitude and acts for only about 140° crank angle. Therefore, in the case of the actual load diagram, the journal would reverse its direction from a smaller eccentricity after the inertia load thereby further increasing the maximum attitude under the gas load.

The importance of this load reversal is best indicated by the observation that unless the load does change in sign no frequency of load variation can produce an attitude other than 1.

It is exactly this fact which makes the problem of the two cycle wrist pin bearing so difficult. Since there is no load reversal, the rotation, small though it may be, must produce the film on which we are to operate.

Discussion by Mayo D. Hersey

U. S. Naval Engineering Experiment Station, Annapolis, Md., and M. I. T., Cambridge, Mass.

It is not clear why Newton's second law is needed in deriving Reynolds' equation. It would also appear that Newton's law of viscous flow is required, rather than a mere defining equation for viscosity. Reynolds' equation assumes that the viscosity, as usually defined, will be independent of shear rate; whereas the definition itself leaves this question open.

May we not adhere to the old terms "eccentricity ratio" or "relative eccentricity" since "attitude" has been used so long by Swift and others to denote the angle between load line and line of centers?

It is hardly clear why the load capacity of a bearing should be proportional to certain factors on the right hand side of Eq. (3) unless we are expected to apply some intuitive form of dimensional reasoning. Turning to Eq. (4), should the last member on right be multiplied by some length factor? Otherwise, since n is a pure ratio, the dimensions do not seem to check.

Can the peak pressure in pounds per square inch, and the piston pin diameter, be stated for the example represented by Fig. 7? The author's solution for the piston pin bearing subject to a sinusoidally fluctuating load is of special interest, showing a six-fold greater load capacity than for the corresponding steady load bearing. However, the absolute value calculated for the eccentricity, namely about one-quarter of the radial clearance, is astonishingly small, thus inviting further consideration of the assumptions involved. How can the assumption of symmetrical positive and negative loads be justified? How would the solution work out for fluctuating loads that are always downward, never reversed? It is hoped that such matters are more fully developed in the forthcoming book by Professor Shaw and Mr. Macks, the publication of which is anticipated with very great interest.

Author's Closure to Discussions

The author agrees with Messrs. Pigott and Cunningham in that the hydraulic bearing analogy will not yield useful results with regard to friction, bearing temperatures, or oil flows. The system presented should only be employed for the purpose intended, namely to estimate qualitatively the influence of various types of loading upon the relative capacities of dynamically loaded journal bearings.

Neither the analogy nor the equations presented take into account inertial effects due to the mass of the oil and the acceleration forces that may result from the sudden reversal of the direction of oil flow under conditions of dynamic load. The inertia forces on a particle of oil are negligibly small except when large bearings operate at rapidly varying rates of speed.

The hydraulic analogy is not limited to cases where the load is fixed in direction as stated in the comment of Messrs. Pigott and Cunningham. In all cases a bearing that is subjected to a rotating load may be replaced, for purposes of analysis, by a bearing operating under a load of constant direction but with altered journal and bearing velocities. This point is illustrated in the interesting example of Messrs. Pigott and Cunningham in which a bearing with journal operating at speed N is subjected to a load of constant magnitude that rotates in the direction of the journal at a speed N/2 and is found to have a zero load capacity.

The use of two connected pistons in place of a single piston is a useful extension of the Analogy of Fig. 4. Then, when the load is increased, positive pressure would develop at the crown of one piston while negative pressure would tend to be produced at the crown of the other piston. Inasmuch as fluids cannot act in tension under conditions found in dynamically loaded bearings, the fluid would separate at the negatively loaded piston. This altered picture would emphasize the well known fact that positive pressure is far more significant in a bearing than large theoretical values of negative pressure.

The author is grateful to Messrs. Pigott and Cunningham and Mr. Hersey for drawing attention to typographical errors in equations (7) and (4) respectively. The appropriate changes were made.

The several queries contained in Mr. Hersey's comment will be briefly considered in order. Newton's

second law is applied to an isolated fluid particle as the first step in deriving equation (1). When it is assumed that acceleration or inertia terms may be neglected Newton's second law reduces to one of the equations of statics. Newton's law of viscous flow is the defining equation for viscosity referred to, viscosity being defined as in fluid mechanics as the proportionality factor relating the shear stress and the angular rate of shear in a Newtonian fluid.

The term Attitude is not original with the author, but is to be found in the engineering literature (for example see "Lubrication" by A. E. Norton, McGraw Hill Book Co., Inc., N. Y., 1942). The angle between the load line and the line of centers is frequently called the Attitude Angle to avoid confusion.

The intuitive form of dimensional reasoning referred to by Mr. Hersey in applying equation (3) might be better described as the process of comparing the right hand side of equation (3) for the ordinary journal bearing with that for a dynamically loaded bearing. The right hand side of equation (3) is $(K U_1 \frac{\partial h}{\partial x})$ for the ordinary journal bearing but may change sign with a multiplier K for certain types of dynamically loaded bearings, where K is a constant. The Dynamically loaded bearing can then be thought of as an ordinary journal bearing operating at a speed KU_1.

The diameter of the piston pin of Fig. 7 is 1.44 inches and the maximum unit bearing load is 9400 psi for the conditions of this figure.

As Mr. Bidwell points out the wrist pin of a two stroke cycle engine presents a much more difficult lubrication problem than that for a four stroke cycle engine due to the failure of the load to reverse its direction in the two stroke cycle engine. In this case the oscillatory motion of the piston pin, due to the periodic change in connecting rod angularity, must largely be relied upon for effective lubrication.

The author does not concur in Mr. Bidwell's belief that the journal of Fig. 7 will continue to approach the bearing surface after the peak load has been attained. This would tend to be the case if the process occurred very slowly and equilibrium could be established as with some of the bearings discussed by Mr. Edwards. However, the entire process considered here occurs in a fraction of a second and the very process which gives rise to the squeeze generated pressure in a bearing that is subjected to a suddenly applied load will likewise tend to prevent the fluid pressure from dropping instantaneously when the load is released. It is rather expected that at the speeds considered here the compressibility of the oil will endow the oil film with the elastic properties of a spring, and when the load is suddenly decreased the pressure in the oil film will tend to cause the journal to move away from the bearing surface.

As both Mr. Hersey and Mr. Bidwell point out the symmetrical sinusoidal loading cycle considered in figure 9 does not bear much resemblance to the actual load diagram of Fig. 8 except in the vicinity of the large peak. It was not intended to imply that the loading cycle of Fig. 9 represents an exact equivalent of the actual cycle throughout an entire period but merely that the two curves are similar in the region of the peak load and hence should be expected to cause approximately the same maximum attitude.

The author wishes to express his gratitude to Messrs. Pigott, Cunningham, Hersey, Bidwell and Edwards for contributing their interesting and thought provoking discussions to this paper.

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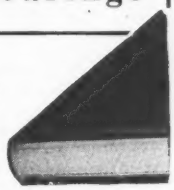

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LABORATORY EVALUATION OF EP GEAR OILS*

by
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Dr. Egbert M. Kipp was born in Angola, Portuguese West Africa, of missionary parents. He graduated from Iowa Wesleyan College in 1934, obtained his A.M. degree from Boston University in 1935, and his Ph. D. degree from Pennsylvania State College in 1939. He joined the staff of the Aluminum Company of America as a member of the Physical Chemistry Division in 1939; was made Assistant Chief in charge of the Oils and Lubricants Section in 1944; and became Chief of the Lubricants Division in 1947.

..He is past president of the American Society of Lubrication Engineers and serves as chairman of several of its national committees. He is very active in community affairs; is a member of the Pittsburgh Section of the American Chemical Society; Chairman of the Speaker's Bureau of that society, and is a member of the Pittsburgh Chemist's Club. Honorary societies include Sigma Xi, Phi Lambda Upsilon, Pi Mu Epsilon, Iota Phi, and Sigma Phi Upsilon.

E. M. KIPP

Introduction

The development of suitable laboratory methods for the evaluation of gear oils would be an important aid in the selection of gear lubricants. The method should be one by which tests can be completed in a matter of hours rather than days and in which the test pieces are of low cost. Test equipment such as the Timken, Falex Tester, Shell Four-Ball Lubricant Tester, and others of this type, though widely employed, did not, at first, appear to offer the solution to the present problem. Chief among the apparent shortcomings was the *reputed* fact that these machines did not rate any given series of lubricants in the same order, and that correlation with field performance characteristics had not always been established.

Because of the emphasis frequently placed by other investigators in this field upon the lack of correlation among various lubricant testing apparatus of the type under discussion, it was felt that as an initial phase of a long range program regarding laboratory methods for evaluating extreme pressure lubricants, it would be advantageous to explore in some detail the degree of correlation or lack thereof in the evaluation of a representative number of commercial heavy duty extreme pressure gear lubricants utilizing the Timken, Shell Four-Ball (Precision Scientific model) and Falex machines. In carrying out this project, forty one proprietary gear lubricants constituting a representative cross section of such materials were selected. It was felt that this would provide a sufficient number to indicate in an exploratory way the probable situation with reference to the matter of correlation of results among various machines. It is to be noted that the investigation covered in this paper is of a purely exploratory nature and that no implications regarding application to field problems are made or intended. The scope of the investigation in essence is only that indicated, namely, an evaluation of forty one representative gear lubricants utilizing the Timken, Falex, and Four-Ball machines with a view toward determining whether there was any agreement whatsoever with regard to the rating of the lubricants among the various machines.

Experimental Procedure

Forty one proprietary gear lubricants were obtained from various lubricant suppliers for this investigation. The physical properties and additive content of these oils are given in Table I. No effort was made in this initial program to segregate the oils into viscosity gradings even though unpublished data obtained in these laboratories had indicated a strong effect of viscosity

* Paper presented before the Pittsburgh Section of A.S.L.E., January, 1949.

TABLE I
PHYSICAL PROPERTIES OF OILS USED IN THIS INVESTIGATION

Oil	Viscosity SSU/100 F.	Viscosity SSU/210 F.	Neut. No.	Sap. No.	Phosphorus	Chlorine	Sulfur	Lead
Code 1	4010	210	.2	5	....	3.5	.2	....
Code 2	2940	134	5	16	....	.1	2.4	3.6*
Code 3	3030	....	.1*	22*	....*	3.9*	1.5*	....*
Code 5	2370	134	2	17	....	.3	2.7	2.3
Code 6	3820	198	.1	10	....	1.9	.4	.01
Code 7	1420	105	2	14	....	1.0	1.2	1.7
Code 8	3900	186	3	19	....	.07	1.4	....
Code 9	3980	198	.3	14	....	2.6	.6	....
Code 10	3320	174	.3	.3	....	.1	2.6	.02
Code 11	3900	169	5	23	....	.1	1.5	1.8
Code 12	1320	100	.2	.2	....	.05	.4	....
Code 13	1040	93.8	.1	7	....	.1	.4	....
Code 14	950	85	.1	20	....	4.1	1.4	....
Code 17	6170	212	4	36	.04	....	1.5	....
Code 18	930	70.7	1	17	....	2.9	.8	....
Code 19	3270	179	.1	24	....	3.1	1.0	....
Code 20	1180	93.0	.7	19	....	3.3	.7	....
Code 21	4670	201	.1	5	....	1.9	.6	.2
Code 22	1300	91.4	4	0	....	.1	3.4	2.0
Code 23	1300	119	1	29	....	.6	2.0	....
Code 24	1260	86.2	.4	15	....	2.9	.6	....
Code 25	7130	305	2	21	....	2.2	.5	2.4
Code 26	8340	329	2	21	.004	2.4	.2	3.0
Code 27	7130	308	.5	13	....	3.3	.6	....
Code 28	7480	324	3	22	....	.2	1.1	1.3
Code 29	966	88.3	.6	3	....	.1	1.8	....
Code 30	1650	122	.5	7	....	.6	2.6	.2
Code 31	3420	145	3	11	....	.2	1.9	1.8
Code 32	4510	168	3	14	....	.2	2.1	2.5
Code 33	945	78.2	4	15	....	.4	.5	3.2
Code 34	3650	139	.7	4	....	.05	1.5	....
Code 35	1130	88.3	5	22	....	.05	1.3	3.6
Code 36	4000	175	5	13	.06	.04	.7	1.6
Code 37	4510	207	.1	3	....	.3	.3	....
Code 38	3920	184	3	4	....	.05	1.4	....
Code 39	1290	108	.7	13	....	....	1.0	....
Code 40	3040	112	2	11	....	.2	2.1	.8
Code 41	1010	88.8	.2	18	....	2.6	.6	....
Code 42	1070	90.2	.1	10	....	2.4	.3	.2
Code 44	456	60.4	2	3	.05	.04	.3	....
Code 46	1310	103	.2	9	....	.6	.9	.03
Code 47	1630	100	3	9	....	....	2.3	2.2

*Estimated

upon performance in some of the test machines. Initially, we desired only to take the oils as they came without further classification with regard to chemical content or physical properties.

Each of the lubricants was evaluated on the Four-Ball, Timken and Falex machines, according to procedures later described. In utilizing these three machines, one observation was made on the Falex machine, namely, the failure load; two observations were made on the Timken machine, these being the Timken OK load and the psi value at the OK load; three observations were made on the Four-Ball machine, these being the Navy pressure wear index value, the 2½ second seizure delay value, and the weld point value. These various individual values will be referred to by the names indicated. When the individual values are combined in the case of any given machine in order to obtain machine ratings, the resultant value will be referred to as the corresponding Timken, Four-Ball, or Falex machine ratings. When individual machine ratings are further combined, the resultant values are referred to as over-all machine ratings.

I. Four-Ball Procedure

The Four-Ball Extreme Pressure Lubricant apparatus employs a system of four steel balls, three in contact with each other in one plane in a fixed, triangular position in a reservoir containing the oil sample, and a fourth ball above and in contact with the other three. The upper ball is rotated at a speed of 1800 rpm by a two horsepower motor during the test. The upper ball can be pressed against the lower three at any desired pressure by applying the correct weight to a lever arm.

Several load bearing determinations may be made using the Four-Ball machine. Determinations used in this study consist of Navy *P-WI*, 2½ second seizure delay, and weld point. Procedures for these various methods will be described briefly.

A. Navy P-WI

The U. S. Naval Engineering Experiment Station instigated a testing procedure for the Four-Ball machine which results in the determination of the so-called Pressure-Wear Index (*P-WI*). This method employs a series of 60 second test runs at various loads using a new set of balls and a fresh sample of oil for each separate run to determine the maximum load (L_b) at which no seizure is recorded on the automatic friction recording device on the machine, and also the minimum load at which instantaneous seizure occurs (L_e)*. The diameters of the scars on the three lower balls resulting from the 60 second trials at these transition

* A no-seizure curve would consist of a smooth flat friction curve. When seizure delay occurs, a sharp rise in friction is recorded at some interval within the 60 second test period. The minimum applied lever load required to cause seizure immediately upon starting the motor is termed the instantaneous seizure load.

loads are measured along diameters parallel to, and in a direction perpendicular to, the striations in the scar area with the aid of a calibrated microscopic eyepiece scale. These diameters are averaged for each of the transition loads, and *P-WI* is calculated from the equation,

$$P\text{-}WI = \frac{L_c - L_b}{D_c^2 - D_b^2}$$

where L_b and L_c are transition loads expressed in kg, D_b is the average diameter (mm) at L_b, and D_c is the average diameter (mm) at L_c.

B. 2½ Second Seizure Delay

It was mentioned above that seizure can occur at any time from less than 1 second to 60 seconds depending upon the applied load. After a series of tests have been made to determine the transition loads, enough data are usually available to plot a curve relating the number of seconds required for seizure to occur at the various applied loads. Interpolation on this curve is then made to find the load necessary to cause seizure to occur at precisely 2½ seconds. This load (kg) is termed the load required for 2½ second seizure delay.

C. Weld Point

Tests are continued at continually increased loads until actual welding of the four balls occurs. This load is the weld point load. It is recorded in kilograms and may be used in evaluating the load bearing capacity of the lubricant under investigation.

II. Timken Procedure

The Timken Tester utilizes the principle of rotating a steel ring against a steel test block. The steel ring, ½ in. wide and having an outside diameter of approximately 2 in., is placed on the mandrel of the machine, the speed of which may be varied from 400 to 800 to 1200 rpm (200, 400, and 600 feet per minute, respectively). All tests run during this investigation were run at a mandrel speed of 800 rpm (400 feet per minute). Pressures between the bearing and the steel test block upon which it runs are regulated by applying a suitable load to a set of lever arms. Frictional force may be measured by applying sufficient weight to an arm provided for that purpose.

A. OK Load

A series of ten minute tests are run using a new bearing and a new test block for each individual test run. The lubricant being tested is flooded onto the bearing throughout the test.

To establish the OK load, it is necessary that the highest load be found which gives at least two out of three tests which do not score, this load being such that the next highest load interval yields at least two scores out of three tests.*

B. OK Unit Load (Psi)

The widths of the wear scars on the test block made by the ten minute runs at the OK load (see Section A) are measured using a calibrated microscopic eyepiece scale. These scar widths are then converted to inches and multiplied by the scar length which is ½ inch (since the bearing is ½ inch wide) to obtain the scar areas. These areas are divided into the pressure at the bearing test block interface which is equal to 10 times the lever load plus approximately 15, the latter term being used to compensate for the weight of the lever arm and the negative force resulting from friction. This quotient gives the OK unit load in pounds per square inch (Psi).

III. Falex Lubricant Tester

The Falex (or Cornell) Tester utilizes a steel test journal rotating at a constant speed of 290 rpm between two steel test bearings each having a V-shaped groove in one of their faces. The journal is held in the rotating shaft by a brass pin in such a way that the journal rotates in the V-shaped notches of the test bearings. Various loads can be applied to the bearings through a special jaw arrangement with a special loading device. The maximum load which it is possible to apply is 4500 pounds at the bearing surface. The unit pressures may exceed 100,000 psi, depending upon the width of the scars produced.

The tester is also equipped with a torque measuring device, making it possible to read the torque directly from a gage calibrated in pound-inches.

A. Failure Load

The determination of the Falex failure load is begun by inserting a new set of test pieces into the machine, and filling the reservoir surrounding the test pieces with about 60 ml. of the oil being tested. The machine is started and a load of 500 pounds is applied for five minutes. This period is called the run-in period. The load is then increased by means of the automatic loading mechanism until failure results. The failure load and the cause of failure are recorded. This procedure is repeated six times for each oil, replacing the test pieces and oil sample each time.

Failure is caused by high friction resulting in the shearing of the brass pin or journal (*F*) or by the welding together of the test pieces (*W*). If failure occurs during the run-in period, a failure load of 500 pounds is arbitrarily assigned for that particular test. The failure loads are then averaged and recorded with the predominating cause of failure being indicated.

Experimental Results

Falex failure load, Timken OK load and psi, and Four-Ball *P-WI*, weld point, and 2½ second seizure delay values for the gear oils tested during this investigation are given in Table II.

Correlation of Results

Investigation of possible correlation among the results reported in Table II was carried out by comparing,

A. Individual determinations.
B. Individual machine ratings.
C. Individual machine ratings against combined machine ratings.

The results obtained are briefly discussed in the following sections.

A. Correlation of Individual Test Data

Correlation results among the individual values obtained on each machine are summarized in Table A. It was emphasized that the term "correlation" in this and subsequent discussions refers to a qualitative correlation only, as indicated by plotting the various results under discussion against one another. In this regard Fig. 1 is typical of what is intended to represent

* In this investigation the OK load was determined to the nearest five pound load increment if the OK load were over 20 pounds, and to the nearest two pound interval if the OK load were less than 20 pounds.

TABLE II

FALEX, FOUR-BALL AND TIMKEN LOAD BEARING VALUES FOR OILS IN TABLE I

Oil	Viscosity SSU/100 F	Viscosity SSU/210 F	Four-Ball Weld Point	Four-Ball 2½" Seizure Delay	Four-Ball Naval PWI	Timken OK Lever Load	Timken OK Unit Load	Falex Falex Failure Load
Code 1	4010	210	125 kg.	99 kg.	9	40#	13700 psi	1300#-W
Code 2	2940	134	220	61	28	40	15500	500-Wr
Code 3	3030	....	290	133	57	25	14600	4000-F
Code 5	2370	134	240	72	41	..	.....	500-Wr
Code 6	2820	198	190	106	16	65	22500	1590-W
Code 7	1420	105	320	92	39	55	32000	2640-F
Code 8	3900	186	220	105	17	40	16100	2050-W
Code 9	3980	198	350	135	76	25–30	15000–16300	3180-F
Code 10	3320	174	260	92	24	30	15800	1730-W
Code 11	3900	169	500	103	37	25	10200	2420-F
Code 12	1320	100	120	60	5	8	9300	620-W
Code 13	1040	93.8	250	116	32	30	13100	2620-F
Code 14	950	85	470	130	79	25	12500	3640-F
Code 17	6170	212	230	100	20	60	33800–36600	2030-W
Code 18	930	70.7	350	83	66	40	19500	3900-F
Code 19	3270	179	340	130	64	25	15200	3600-F
Code 20	1180	93.0	340	109	80	50	27400	3780-F
Code 21	4670	201	240	144	20	40	13400	3180-F
Code 22	1300	91.4	800	...	..	25	7700	2940-F
Code 23	2240	119	200	103	9	20	11700	1920-W
Code 24	1260	86.2	280	100	37	25	13200	1500-W
Code 25	7130	305	330	105	165	35	13200	3220-F
Code 26	8340	329	300	103	87	40	17400	2640-F
Code 27	7130	308	360	115	175	60	20300	2980-F
Code 28	7480	324	330	127	320	20	11200	1730-F
Code 29	966	88.3	210	97	15	..	.....	1510-W
Code 30	1650	122	270	100	42	..	.....	1580-W
Code 31	3420	145	250	78	39	45	13800	1080-W
Code 32	4510	168	210	98	32	40	15400	500-Wr
Code 33	945	78.2	280	86	38	40	11000	3100-F
Code 34	3650	139	290	90	30	30	14600	2030-F
Code 35	1130	88.3	410	82	48	50	13100	1180-W
Code 36	4000	175	210	105	28	40	14800	500-Wr
Code 37	4510	207	...	...	..	14	9120	500-W
Code 38	3920	184	190	102	22	40	17600	2080-W
Code 39	1290	108	230	100	22	30	16500	1970-W
Code 40	3040	112	190	77	29	30	10800	550-Wr
Code 41	1010	88.8	350	93	61	30	15900	3580-F
Code 42	1070	90.2	210	84	23	40	15500	1550-W
Code 44	456	60.4	150	...	.4	12	10800	1640-W
Code 46	1310	103	330	97	49	25	11600	3030-F
Code 47	1630	100	210	65	29	35	12700	500-Wr

qualitative correlation, while Fig. 2 represents what is considered to be a case showing no correlation whatsoever.

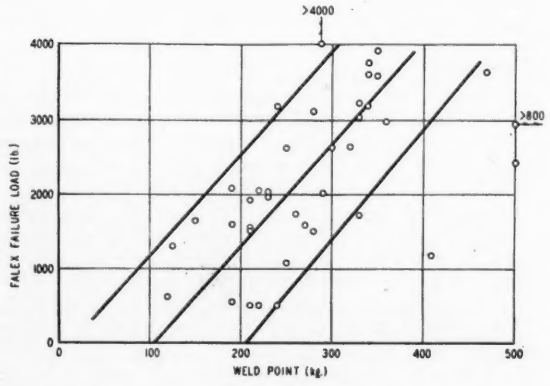

Fig. 1

Relation Between Four-Ball Weld Point Values and Falex Failure Loads

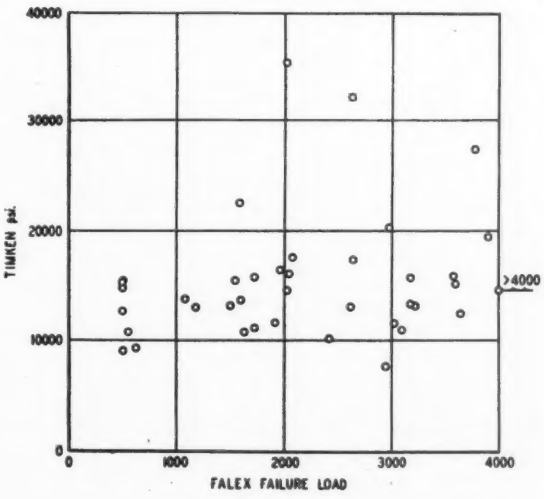

Fig. 2

Relation Between Timken PSI and Falex Failure Load Values

TABLE A

CORRELATION CHART—INDIVIDUAL TEST DATA

	Falex	Timken OK	Psi	P-WI	4-Ball 2½" Seizure Delay	Weld
Falex	X	No	No	No	No	Yes
OK	No	X	Yes	No	No	No
Psi	No	Yes	X	No	No	No
P-WI	No	No	No	X	No	Yes
2½" Seizure Delay	No	No	No	No	X	No
Weld	Yes	No	No	Yes	No	X

B. Correlation of Individual Machine Ratings

In order to obtain a comparison between the various machine ratings, it was necessary to first compute the ratings for each individual machine. This was done by listing the various individual determination values of the oils given in Table II in decreasing numerical order and assigning a rating of 1 to the oil giving the highest results for each determination, 2 to the second highest rating oil, etc. These individual ratings were then grouped according to the machine from which they were obtained and the totals for each oil computed for each machine. For example, an oil ranking 1 on *P-WI*, 4 on the 2½" seizure delay, 8 on the weld point determination, would receive a total combined rating value of 13 on the Four-Ball machine. The totals obtained in this manner for all of the individual oils were then rearranged according to decreasing merit, a ranking of 1 being assigned to the highest rating oil, 2 to the next best oil, etc. Table B summarizes the degree of correlation noted among the various individual machine ratings.

TABLE B

CORRELATION CHART—INDIVIDUAL MACHINE RATINGS

	Four-Ball	Falex	Timken
Four-Ball	X	Yes*	No
Falex	Yes*	X	No
Timken	No	No	X

*See Fig. 3.

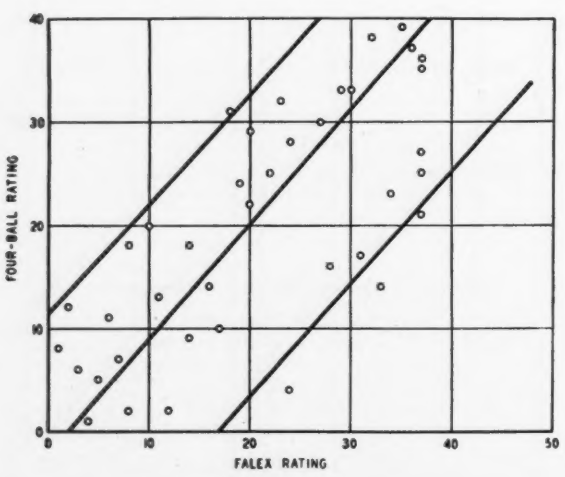

Fig. 3

Relation Between Four-Ball and Falex Ratings

C. Individual Machine Ratings Against Combined Machine Ratings

A combined over-all rating was obtained by adding Four-Ball, Timken, and Falex ratings for each oil, rearranging, and rerating, giving the oil with the lowest total (best performance) a rank of 1, etc. The correlation results are sumarized in Table C.

TABLE C

CORRELATION CHART—INDIVIDUAL MACHINE RATINGS VS. COMBINED MACHINE RATINGS

	Over-All
Falex	Yes*
Four-Ball	Yes
Timken	No

*See Fig. 4.

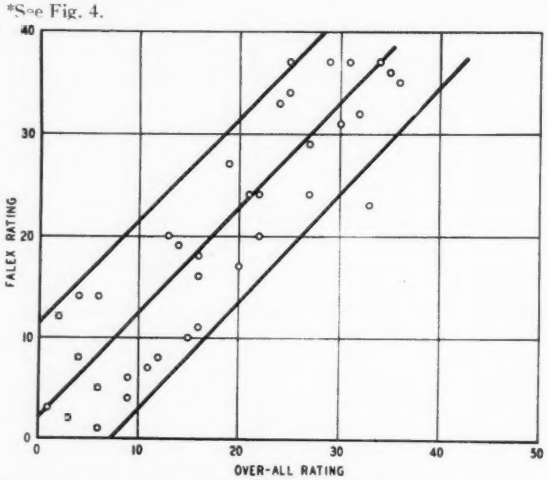

Fig. 4

Relation Between Falex and Over-All (Combined Falex, Timken and Four-Ball) Ratings

General Discussion

It is apparent from the data herein reported that limited correlation does exist between Falex machine and Four-Ball machine ratings. The fact that even the amount of correlation noted was found to exist is all the more significant in view of the fact that the oils were tested as a unit and that no attempt was made to further classify the oils according to viscosity groups, type of additive employed, etc. As indicated elsewhere in this paper, other investigations in these laboratories have indicated that the Timken machine is extremely sensitive to viscosity variations within the viscosity ranges represented by the lubricants employed in this investigation. There appears, therefore, to be some reason to believe that further investigation of the effect of these variables upon the manner in which each machine rates a lubricant may be expected to further clarify the present confused picture as to why different lubricants appear to perform differently in various types of lubricant test apparatus.

It is also significant to note that utilization of either the Four-Ball or the Falex machines alone would apparently have given essentially the same general relative rating as would have been obtained by the combined use of the Falex, Four-Ball, and Timken machines. It is also of interest to note that the Timken machine ratings do not appear to correlate in any obviously direct way with either over-all ratings or with individual Falex and Four-Ball ratings. This again may be due to the greater sensitivity of the Timken machine to residual viscosity effect. This matter is being investigated further.

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W. A. ZISMAN

NON-FLAMMABLE HYDRAULIC FLUIDS[1]*

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ABSTRACT

In addition to the properties normally required in hydraulic fluids such as lubricity, stability, adequate liquidus range, inertness to metals and packing materials, there is a need for a less flammable fluid to care for the fire hazards that are inherent in military and certain commercial hydraulic applications. The common flammability tests, spontaneous ignition temperature, flash and fire points were found to be incapable of predicting the relative flammabilities of fluids to the variety of fire hazards to which they may be exposed in service. Two new tests were developed, the incendiary fire and spray flammability tests, to evaluate the resistance of fluids to other hazards.

The relative flammabilities of a variety of fluids including petroleum oils, carboxylic and phosphate esters, glycols and derivatives, silicones, chlorinated and fluorinated compounds, and water-base fluids were studied using these methods. It was concluded that the hydrocarbon and carboxylic ester fluids were quite flammable. Increasing the proportions of oxygen, phosphorus and halogen constituents and the concentration of water in a fluid composition reduced the flammability. The action of fire-resistant additives like volatile "snuffers" and non-volatile "quenchers" is imperfectly understood and the need for physicochemical research on this problem is evident. It is important to understand the effect of high-boiling liquids in the hydraulic fluid range on elastometers as such information will serve as a guide in research and development on new fluids and suitable packings.

It is concluded that only three hydraulic fluids have been developed which are sufficiently flame-resistant: Hydrolubes, the hexachlorobutadiene-base fluid "A," and the chlorinated aromatic base fluid "PS." Only Hydrolube can be used with present packings. Development of Hydrolube U-4 is now under way to improve the freezing point, V.I., and anti-wear properties. Both chlorinated fluids will require the development of new packing materials for hydraulic use, the problem being more difficult for the chlorinated aromatic type.

Introduction

When hydraulic systems are mentioned it is usual to think of such familiar devices as hydraulic jacks, lifts and presses, automotive brakes and shock absorbers. But there are a host of other applications such as machinery for pipe bending, plastic, and metal molding, stamping, die-casting, printing, bottling, filling, hoisting, grinding and milling. In the armed services wide use is made of hydraulic systems to operate gun turrets; control gun fire; hoist anchors, navigate ships and planes; handle, launch and land planes on carriers; open bomb bays; fold wings and retract aircraft landing gear, etc. The development of improved hydraulic fluids is, therefore, of much interest to industry as well as the Navy.

The properties required of hydraulic fluids include: (1) low compressibility; (2) adequate lubricity over the range of loads, temperatures and speeds to be encountered; (3) freedom from deleterious effects on packings and plastic parts; (4) non-corrosiveness to metals like iron, steel, copper, bronze, aluminum and duralumin which are in common use in hydraulic machinery, and to the iron, tin, terne plate and solder used in standard shipping and storage containers; (5) adequate corrosion inhibition to prevent rusting caused by exposure to humidity and accumulation of water of condensation; (6) adequate thermal and chemical stability; (7) low foaming tendencies; (8) minimum flammability; (9) good demulsibility; and (10) freedom from health hazards.

Water was the earliest hydraulic fluid, and its low compressibility, inertness to packings, non-toxicity and

[1]The opinions or assertions contained in this paper are the authors' and are not to be construed as official or reflecting the views of the Navy Department.

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non-flammability are still desirable properties. Various aqueous solutions were introduced later, the more common being mixtures of glycerine or glycol with water (1, 2). As hydraulic equipment became more complex, difficulties with corrosion, inadequate lubricity and narrow operating temperature range necessitated the introduction of other fluids. Today petroleum-base fluids constitute a high percentage of hydraulic fluids because they are cheap, non-toxic and available in a wide range of viscosity grades. Petroleum hydraulic oils in common use have viscosities at 100 F ranging from 10 to 260 centistokes (cs). By properly refining and selecting petroleum fractions, fluids may be prepared (3, 4) having little tendency to form gums and sludges under common operating conditions. Additives have been developed (5) to decrease wear, improve oxidation stability, prevent rusting caused by water (6, 7) and reduce foaming (8, 9). Finally, where necessary to obtain high V.I. fluids, oil-dispersible linear polymers such as polybutenes (10) and polyesters (11) (15) have been developed which, when added in minor concentrations, increase the viscosity and improve the viscosity-temperature characteristics. The wide variations in petroleum-base hydraulic oils are exemplified by the requirements of some military specifications given in Table I.

TABLE I

TYPICAL REQUIREMENTS OF U. S. GOVERNMENT SPECIFICATIONS FOR PETROLEUM HYDRAULIC OILS

	Navy Department Specification 14-0-19		14-0-15	*Bureau Ordnance Specification*		*U. S. Army Specification*	*Air Force-Navy Aero. Specification*
	Grade 2075*H**	*Grade* 2110*H**	*Grade* 2190*T**	51-*F*-21	51-*F*-23	*No.* 2-79*B*	*AN-O*-366
Union Color No. (Max)	Determined	Determined	Determined	2	4	4	Required (c)
Reaction	Neutral	Neutral	Neutral	....	....	....	....
Neutralization No. (Max)	0.10	0.10	0.10	0.3	0.3	....	0.20
Precipitation No. (Max)	None	None	None	0.05	0.05	0.05	0
Copper Corrosion at 212 F	None	None	None	None	None	None	None
Sulfonates	None	None	None	....	....	....	....
Carbon Residue, % (Max.)	0.10	0.20	0.40	....	....	....	....
Ash, % (Max)	Determined	Determined	Determined	....	....	....	....
Water, % (Max)	None	None	None	None	None	None	None
Total Sulfur, % (Max)	0.50	0.50	0.50	....	....	....	....
Flash Point, F (Min.)	315	325	350	225	300	225	200
Fire Point, F (Min)	....	....	....	235	325	....	....
Cloud Point, F (Max)	....	....	....	−35 (a)	−35 (a)	....	−65 (a)
Pour Point, F (Max)	Determined	Determined	35	−40	−40	−50	−75
Viscosity, cs. at 210 F (Min)	....	....	....	10	8	10	....
130 F (Min)	13.1–18.2	18.2–25.2	39.8–44.2	....	....	....	10
100 F (Min)	....	....	....	27	40	38	....
0 F (Max)	2180	4360	....	215	1500	....	....
−25 F (Max)	....	....	....	600	7500	....	....
−40 F (Max)	....	....	....	....	....	7500	500
Viscosity Index	75	75	....	....	....	....	....
Emulsion Test	Required	Required	Required	Required (b)	Required (b)	....	....
Rust Inhibition	Required	Required	Required	Required	Required	Required	....
Oxidation Stability Test	Required	Required	Required	Required	Required	Required	Required
Foaming Test	Required	Required	Required	....	....	Required	....
Shear Stability Test	....	....	....	Required	Required	Required	Required
Wear Tests	....	....	....	Required	Required	Required	Required
Evaporation Test	....	....	....	Required	Required	....	Required
Rubber Packing Tests	....	....	....	....	....	Required	Required

*Oils of same viscosity grade and approximately the same characteristics, but without rust inhibition properties are also specified.

(a) 72-hour storage test.
(b) Water sludging test.
(c) Comparison with a dyed oil sample.

Castor oil-base hydraulic fluids (12) were widely used before the war in brake and shock absorbers. They are generally prepared by diluting either refined or blown castor oil with organic solvents and are stabilized with antioxidants and corrosion inhibitors. Their principal advantages are freedom from deleterious effects on common packings, water tolerance, long life and low pour points. Specifications by the S.A.E. (13) now exist for such fluids for automotive and aeronautical uses. In many applications they are being replaced by new fluids developed during the war which are based on the use of polyglycol ethers plus diluents and inhibitors (12). Automotive uses alone require in the U.S.A. an annual production of 3,000,000 gallons of hydraulic fluids.

A thorough study of the suitability of any new hydraulic fluid is necessary but time-consuming and expensive. Determinations of the chemical and physical constants of the fluid, the toxicity, and the effect on packings are generally made in the laboratory. The last requirement is especially critical or significant, e.g., special synthetic rubber packings (14) were required to permit the use of the petroleum-base hydraulic fluids developed under the auspices of the O.S.R.D. (15) for aeronautical applications. The laboratory evaluation is followed by "bench tests" to determine the performance characteristics of the fluid in service-simulating equipment. If the bench tests are satisfactory, final tests are made in service equipment or in "mock-ups" which are laboratory installations of standard operating components arranged as used in service.

Flammability of lubricating and hydraulic fluids has long constituted a hazard in specialized industrial and military equipment. Military hazards increased with the adoption of high velocity and incendiary ammunition. Structures of ships, vehicles, and planes now contain numerous controls, actuators, valves, and tanks in addition to a maze of low and high pressure hydraulic lines which increase the probability of fire or explosion from missiles or from the occurrence of leaks in the vicinity of hot engines, exhaust pipes, or superheated steam pipes. Because of the adoption of the lower boiling petroleum-base fluids thickened with polymers, which satisfied the demand for a fluid capable of op-

erating over wide ranges of temperature, the fire hazards are even more acute. Fires and explosions due to hydraulic fluids were significant sources of casualties and losses of equipment on all fronts during the war. Following the war a number of unfortunate commercial and military aircraft crashes and the accompanying fires concentrated interest in minimizing all aircraft fire hazards. In industrial practice, too, where hot equipment or parts are handled by hydraulic machinery such as in steel mills, die casting shops, bakeries, etc., there has been an increasing demand for eliminating the fire and explosion hazards caused by sudden leaks or component failures.

Few fluids in the desired liquidus range are truly non-flammable. Nearly all organic liquids pyrolize (or oxidize) if exposed long enough to temperatures over 500 F, and this is inherent in the strength (or activity) of the chemical bonds. Volatile pyrolytic or oxidation products are often formed which are flammable. The necessity for determining what should be considered as a practical non-flammable fluid was recognized in 1942. It was agreed that in the search for truly non-flammable liquids two other types of fluids should not be overlooked: first, those which although flammable under extreme conditions would not support combustion when the source of ignition was removed, and second, those which though flammable were still so much less hazardous than available fluids as to deserve attention as a less flammable fluid, or as a component of such a fluid.

Since a wide variety of experimental methods have been considered for observing or measuring the flammabilities of liquids, a search was made for the minimum number of tests capable of simulating the fire hazards of greatest concern. Standard flammability tests for determining flash point, fire point and the spontaneous ignition temperature (16) (abbreviated hereafter to S.I.T.) though yielding valuable information on relative fire hazards of the fluids were insufficient, and incapable of representing their flammabilities under other conditions. Therefore, a new laboratory method was developed (17, 18, 19) for measuring the flammability of a fluid under empirically controlled conditions in terms of the volume percentage of oxygen necessary in an oxygen-nitrogen mixture to permit an arc-ignited flame to propagate along a metal tube filled with a finely dispersed mist or spray of the test liquid. The apparatus is shown in Figure 1 just after an oil mist explosion with flame and smoke issuing from the open end of the metal tube. A high V.I., low pour point, petroleum-base, hydraulic oil was explosive with an oxygen concentration of 12%, while a highly halogenated hydrocarbon required a minimum of 60% oxygen. Obviously, the first fluid could form explosive mists in the air since air contains approximately 21% of oxygen. The minimum oxygen concentration required to form an explosive mixture will be designated, hereafter, as the "oxygen demand."

As a means of verifying the conclusions obtained with the spray flammability apparatus and also because of their practical significance in connection with the combat use of hydraulic fluids, tests were made on the ability of 50 calibre incendiary bullets to ignite each proposed fluid. These tests were made in cooperation with the Naval Ordnance Proving Ground at Dahlgren, Virginia. High speed Kodachrome motion pictures were taken to record the results. Steel baffle plates were used to insure adequate fragmentation and ignition of the incendiary bullets before impact with the oil container. This work confirmed the predictions based upon the laboratory flammability tests; for example, a

Fig. 1
Spray Flammability Apparatus

one gallon tin can of aircraft hydraulic fluid Specification AN-0-366 exploded from the impact of the incendiary bullet creating a flame front which spread over an area extending for a radius of 20 feet around the can. To illustrate these remarks a photograph of such an explosion is shown in Fig. 2.

All five flammability methods (Flash and Fire Points; S.I.T.; Spray Flammability and Incendiary Fire) were applied to the study of a wide variety of hydraulic fluids, synthetic oils, and organic liquids of promise (18, 19, 20). It is noteworthy that fluids having an oxygen demand of less than 40 to 50% in the spray flammability test proved explosive in the incendiary fire test, while those requiring appreciably more than 50% of oxygen did not explode at all.

From the investigation of the flammability of liquids, a number of interesting conclusions were reached. No available petroleum-base or castor oil-base hydraulic fluid was sufficiently flame-resistant. The addition of such antioxidants as diphenylamine and tetraphenyl lead, in high concentrations, did not sufficiently improve the resistance to all types of fire hazards studied. The use of volatile "snuffers" or non-volatile "quenchers" dissolved in petroleum oils did not appear to be a promising method of reducing the flammabilities of a fluid because of the large concentrations required. Petroleum oil mists explode readily in the atmosphere and conditions favoring explosions were created at pressures as low as 25 psi with all high V.I. hydraulic fluids in use and oils below the SAE 30 viscosity grade. Few liquids were resistant to all the laboratory tests for flammability, although many were resistant to one or more tests. The most serious fire hazard found was reproduced by the spray flammability apparatus, and the results showed that the ignition of a mist or fog of lubricant in the atmosphere (such as could occur following a break in a high-pressure hydraulic line) can become the source of an explosion as serious as dust explosions in coal mines and grain elevators. These results led to the prediction in late 1942 that significant combat aircraft losses would result from hydraulic explosions. The correctness of this prediction was confirmed by subsequent aircraft losses and the need for aeronautical non-flammable hydraulic fluids was recognized. A similar need exists for non-flammable fluids in high pressure hydraulic systems of ships, especially war vessels.

Fig. 2
Fire and Explosion from One Gallon of AN-0-366 Hydraulic Fluid

Discussion of Fluids Studied

Since no known treatments or acceptable addition agents were known which adequately improved the fire resistance of petroleum-base or castor oil-base fluids, the investigation was extended to numerous other organic chemicals and their mixtures.

A. Esters of Carboxylic Acids

Branched-chain aliphatic diesters were studied early in the war in connection with the development of low volatility, low pour, high V.I. synthetic lubricants for use over the temperature range—80 F to 300 F (21, 22). As some diesters in the desired viscosity and pour point ranges were much less volatile and had higher flash and fire points and S.I.T.'s than comparable hydrocarbons they appeared promising as less flammable fluids. However, the oxygen demand differed little from that of petroleum oils of equal viscosities and incendiary fire tests confirmed these conclusions (18, 19). These fluids also seriously swell aircraft specification (14) rubber packings.

B. Organic Phosphates

Because of their high flash points and reputed flame resistance, the flammabilities of a variety of the available phosphate esters were studied (19). Aliphatic phosphates like trimethyl phosphate and hexamethyl tetraphosphate which have high phosphorus to carbon ratios were significantly fire-resistant since the oxygen demand was 47% and 31% respectively and the S.I.T.'s were approximately 900 F. Unfortunately, the former compound has a flash point of only 225 F while the latter is readily hydrolyzed to form a corrosive acid. Both liquids attack the common types of rubber packings. Entirely aromatic substituted phosphates like tricresyl phosphate, which has been used as the hydraulic fluid in die-casting machines in recent years (23), are more fire-resistant and stable, but they attack packings and have low V.I.'s. The commercial tricresyl phosphates are high-boiling liquids (B.P. 275 C to 290 C at 10 mm.) having viscosities at 100 F of approximately 26 cs., flash points approaching 500 F, S.I.T.'s above 1.000 F. and oxygen demands varying from 19% to 85%. The pour points vary with the concentration of the para isomer and the volatile impurities present, but generally range from —10 F to —15 F. Other phosphate ester base (24) lubricants have been proposed for various specialty applications.

Late in the war two less flammable aircraft hydraulic fluids were developed industrially and submitted to this laboratory. One fluid consisting of a mixture of an organic phosphate and a dicarboxylic ester and additives is identified as fluid "N." The other fluid is a mixture of an organic phosphate and a polyglycol ether (discussed under the next heading—Glycols and Derivatives). Still more recently another phosphate ester base aircraft hydraulic fluid, identified as DMX-8, was described (25) and released for commercial use. The properties of these two fluids are given in Table II. Fluid "N" conforms to the viscometric and low temperature requirements of specification AN-O-366, while DMX-8 will only conform to the minimum requirements of SAE Aeronautical Material Specification 3150. Both fluids will flash and burn when heated, "N" being the more flammable under these conditions. It likewise has the lower S.I.T. but is much less flammable to this hazard than the current aircraft specification petroleum hydraulic oil. But fluid "N" is much less flammable in the spray test, the "oxygen demand" being 54% as compared to 23-25% for DMX-8. The high oxygen demand of fluid "N" indicates that it is not likely to be ignited by incendiary fire but it is probable that DMX-8 would be quite flammable to this hazard. When the spray test was run with the fluid at 100 F instead of 77 F, DMX-8 was flammable with air (oxygen demand 21%) therefore, it will not always be less flammable to this hazard than the petroleum aircraft hydraulic oil in use.

The rust inhibition properties of these fluids were evaluated by the "Static Water Drop" test (7) with distilled water at 140 F. They were found to be superior to those of AN-0-366 petroleum oil in this respect but the metal test specimens were stained where water came in contact with them. Presumably staining was caused by corrosive products resulting from hydrolysis. Some indication of the corrosiveness of the bulk fluid to the common metals was obtained by measurements of weight loss and by the appearance of the test specimens after 72 hours immersion in the fluid at 212 F. The metals immersed in fluid "N" showed no evidences of corrosion in 72 hours but on standing ten days at room temperature after the test the copper and Dow Metal H specimens were corroded. Probably this was also due to hydrolytic products. The DMX-8 fluid showed no evidence of corrosive action on copper, steel, aluminum or cadmium plated specimens in 72 hours but the magnesium alloy specimens were etched and pitted. Both fluids are miscible with petroleum hydraulic oils. Unfortunately, they also seriously attack the 0-ring rubber packings used in existing equipment, and their adoption would require the replacement of nearly all existing packings. Fluid DMX-8 is reported (25) to have demonstrated excellent anti-wear characteristics and a satisfactory shear stability in hydraulic bench tests. No information is available as to its tackiness, foaming tendencies or oxidation stability. In an attempt to distill it under reduced pressure (5 mm) it cracked with the formation of considerable amounts of acid phenols. No information is available as to the performance of fluid "N" in hydraulic bench tests, or of either fluid in aircraft systems.

More suitable nonflammable fluids or fire-quenching diluents can be expected from synthesis and research on compounds containing high proportions of phosphorus. Low molecular weight organic polymers of phosphorus with short alkyl side-chains to give low freezing points and high boiling points appear particularly promising.

TABLE II

COMPARISON OF SOME PROMISING AIRCRAFT HYDRAULIC FLUIDS

Base Fluid Identification	*Petroleum Oil AN-0-366*	*Glycol-Water 51-F-22(Aer)*	*Phosphate Ester DMX-8*	*Phosphate Ester "N"*	*Phosphate Ester H#*	*Hexachlorobutadiene "A"*
Viscosity, cs. at 210 F	5.1	4.0	3.85	5.20	4.64	2.95
130 F	10. (Min)	10.	9.5	10.3	9.7	5.88
100 F	14.	17.	15.0	14.7	13.9	8.54
0 F	90.	245.	285.	108.	108.	75.
−20 F	165.	610.	950.	195.	186.	160.*
−30 F	250.	1000.	2256.	290.	275.	240.*
−40 F	500.(Max)	1850.	6000.*	442.	404.	425.
Pour Point, F	−75 (Max)	−60	<−60	<−70	<−70	<−65
Flash Point, F (C. O. C.)	200 (Min.)	None	360	305	c.a.300 (a)	(a)
Fire Point, F (C. O. C.)	215	None	470	365	360	(b)
S. I. T. (e)	c.a.500	850	>1300	887	795	>1300
Oxygen demand, %	12	>65	23-25	54	55	>85
Density, 68 F	0.80–0.85	1.06	1.08	0.98	1.06	1.5
Packing Materials (c)	Satisfactory (d)	Satisfactory	Poor	Poor	Poor	Poor
Wear Prevention	Satisfactory	Fair	Good	.....	.....	Good
Shear Stability	Fair	Excellent	Good	.....	.....	.....
Hydrolytic Stability	Satisfactory	Satisfactory	Doubtful	Doubtful	Doubtful	.....
Corrosiveness (72 hr at 212 F)						
Cold Rolled Steel	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Satisfactory
Copper	Satisfactory	Satisfactory	Satisfactory	Satisfactory (f)	Poor	Satisfactory
Aluminum 52SO	Satisfactory	Satisfactory	Satisfactory	.....	Satisfactory	Satisfactory
Duralumin 24ST	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Satisfactory	Satisfactory
Cadmium-plated Steel	Satisfactory	Satisfactory	Satisfactory	.....	.....	Satisfactory
Dow Metal C	Satisfactory	Poor	Poor	.....	.....	Satisfactory
Dow Metal H	Satisfactory	Poor	Poor	Satisfactory (f)	.....	Satisfactory
Tackiness	Satisfactory	Satisfactory	Satisfactory	.....	.....	Satisfactory
Surface Tension (Dynes/cm)	28	31	.....	.....	.....	32
Foaming Test	Satisfactory	Satisfactory	.....	.....	.....	Satisfactory
Rust Inhibition Test (g)	Poor	No rust	No rust (h)	No rust	Poor	No rust (h) (i)
Oxidation Stability	Satisfactory	Does not apply	.....	.....	.....	.....
Miscibility with Petroleum Oils	Complete	Immiscible	Complete	Complete	.....	Complete

Base fluid—no additives.
*Extrapolated.
(a) Does not have a true flash point, burns around flame source.
(b) Fluid foams and boils at approximately 385 F.
(c) AN-P-79 rubbers
(d) Present packings designed for this fluid.
(e) Method of Sortman, Beatty and Heron (16).
(f) Corroded metals after standing 10 additional days at room temperature.
(g) Static water drop test, distilled water at 140 F.
(h) Test specimen stained at points of contact with water.
(i) Precipitates formed.

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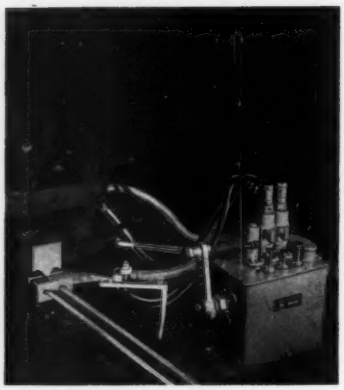
This article will be concluded in the December issue of **"Lubrication Engineering"** along with the entire Bibliography.

NEW PRODUCTS and EQUIPMENT



SPRAY LUBRICATOR

An automatic force-feed spray lubricator which intermittently sprays punch and shear operations with oil was announced recently by Herbert H. Roosa, vice-president of Manzel, Inc.

Test installations in several heavy production plants have made impressive records in increasing punching and shearing output by greatly lengthening the life of dies. Lubrication is claimed to be more thorough and produces cleaner working and operation conditions. Oil consumption is drastically reduced.

Synchronized directly with the machines, the Spray Lubricators force automatically timed jets of oil mists directly onto the punches, shearing edges, dies or other parts. Compressed air and oil are fed simultaneously into a sprayer nozzle aimed at the critical area. The air stream likewise has a cooling effect upon die surfaces.

The new Manzel Spray Lubricators can be used for large or small operations.

In business for more than fifty years as a manufacturer of automatic lubricators and chemical feeders, Manzel plans immediate production of the new device on a quantity scale.

A booklet illustrating and describing the Spray Lubricator is ready for distribution. Write: Manzel, Inc., Buffalo, N. Y.

GRAPHITE FILM

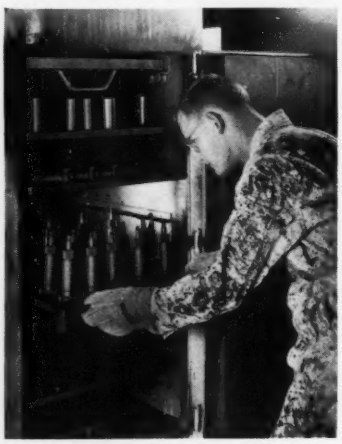
A highly successful means of applying a stable graphite film to practically any surface has been developed after 10 years of research by the Electrofilm Corp., 7116 Laurel Canyon Boulevard, North Hollywood, Calif.

The new stable graphite film, as applied by the Electrofilm process, is extremely thin (0.00015 to 0.0005 in.) but has excellent resistance to abrasion and exceptional bearing strength. The adhesion to the intended surface is high, and on metal and most other surfaces sufficient diffusion of the graphite into the surface is obtained to insure presence of a graphite surface even when the external coating is apparently removed.

The Electrofilm graphite process is applied by spray or dip, following normal surface preparation and has been successfully used on metal, plastics, rubber and ceramics. For most purposes, no plating is required, although Electrofilm graphite can be applied over plated parts.

Electrofilm is stable, and provides the intended surface characteristics over temperatures ranging from —120 F. up and some variations have been tested satisfactorily to 2000 F. The graphite film is unaffected by exposure to solvents and weather.

Cost of applying the new Electrofilm graphite is inexpensive, and the process is licensed to manufacturers or can be applied by the Electrofilm Corp.



ACIDITY TESTING

The neutralization number (acidity) of transformer, turbine and other mineral oils can now be checked reliably and quickly and without technical skill, by means of an inexpensive new portable testing kit, introduced by the Gerin Corporation of Red Bank, New Jersey.

An important feature of the kit is that the neutralizing KOH is in single measured doses, in *sealed* ampoules, instead of being applied from an open bottle. Thus the measure is accurate and the KOH does not lose its strength from contact with the atmosphere. Ampoules are supplied with a neutralization value of .3, .6 and 1.5 mg of KOH per gram of oil.

The test is made with one-third ounce of oil sample and takes about two minutes to complete.

Use of acidity tests is well-established as a quick indicator of oil deterioration. While they are not

the sole criterion of oil quality, they are useful as a screening test to show the need for more elaborate examination.

Furnished are four glass cylinders for making the tests, two pints of a neutral solution of water-alcohol-pink indicator, 20 ampoules of the KOH, and files for opening the ampoules. Solutions, ampoules, equipment and simple instructions are neatly boxed in a handsome hand-rubbed mahogany case . . . a complete self-contained acidity test kit.

For further information, write: The Gerin Corporation, Red Bank, New Jersey.

NEW GULF SHOP

Outstanding among the recently expanded industrial laboratories and machine shops is the new and modern machine shop of the Gulf Oil Corporation's research laboratories at Harmarville, Pennsylvania.

Planned to function 100% as a "tool" for the oil firm's research and exploratory work, the shop is supplying all metal-working requirements of the laboratories' staff of chemists, physicists and engineers.

It also produces most of the parts required for geophysical prospecting equipment made by Gulf and used in its exploration activities throughout the world.

Products of the Gulf shop probably are more diversified than those of any similar shop. They range from clocks for bottom hole pressure gauges used in oil well drilling and production, to harvesters used to gather pyrethrum daisies for use in the company's insecticides. Among notable "firsts" constructed here are the airborne magnetometer, used in geophysical prospecting; the Azon and Razon bombs, guided missiles developed for the last war; and marsh buggies capable of traveling on hard land, in mud, or in water, and currently employed in oil exploration work in Louisiana and Texas. Continuing this impressive list is a percussion drill, the modern prospecting gravimeter, and the Gulf two-cylinder test engine.

Products developed by the laboratories' Lubrication Division are evaluated here in both normal and special test machining operations. Operators who do such test work are especially trained as both machinists and oil test men, and all evaluations are run under the supervision of lubrication research engineers.

The new Gulf machine shop has 18,479 square feet of floor space. The main shop and assembly floor occupy 10,676 square feet, while storage space in the basement and mezzanine consumes slightly more than 2,500 square feet. A grinding room, where a large part of the laboratories' testing of thread-grinding oils is conducted, is partially sound-proofed and is included in the row of special shops. Welding and sheet metal work, including the manufacture of special truck bodies needed for seismograph operations, is carried out in another soundproof room, next to the assembly bay. The stock room carries some 2,600 different items, ranging from waxed paper to steel bars measuring six inches in diameter.

The metal-working machines of the shop include 24 lathes, 13 milling machines, 6 presses, 12 grinders, a thread grinder, 9 vertical drills, a planer, 2 shapers, a gear shaper, a horizontal-boring machine, a jig borer, an engraving machine, a nibbling machine, a spot welder, 2 arc welding units, 2 power shears, a brake, 5 metal saws, and miscellaneous small machine tools.

Between 3,000 and 4,000 jobs will be turned out in this shop in the course of a year.

NEW LUBRICATOR

Newly developed for application where space limitations prohibit the installation of the large lubricators, the type LPM lubricator is a small, compact, sump-mounted unit allowing the engineer a good deal of freedom in his design.

This unit has a wide field of application where closely controlled but infrequent oil feed meets the operating requirements of a machine. A relatively inexpensive unit, it appeals directly to the machine designer who has a small group of bearings to be oiled and a limited budget with which to do the job.

The lubricator is the piston-type unit with an efficient filter to prevent foreign particles being drawn into the distribution system. The action of the lubricator is initiated by simply withdrawing the pull button upon release of which the delivery of a measured quantity of filtered oil to all bearings is carried out automatically.

Used in conjunction with the Bijur small-capacity distribution system, the LPM lubricator is recommended for use—

1. Where small but carefully measured quantities of oil must be fed.
2. Where space limitations are a contributing factor.
3. Where, for improved appearance, a sump-mounted lubricator is indicated.
4. In isolated units involving a group of bearings which cannot be connected to the main lubricating system of a machine.

It is adaptable to any machine layout and affords full opportunity to maintain all standards of form, convenience, efficiency and appearance which have been set for the job. For further information, write Bijur Lubricating Corporation, Long Island City 1, N. Y.

DRY LUBRICANT

A new and less expensive grade of MOLYKOTE lubricant, known as Type Z, has been made available. This new lubricant is a specially prepared, smooth-textured molybdenum compound powder whose treatment and composition is said to render it peculiarly adaptable for extreme bearing pressure applications.

MOLYKOTE, Type Z, has exceptional chemical and thermal stability and inertness, it resists the attack of all but a few acids, and its lubricating characteristics are not affected by temperature from the sub-zero range up to 750 F. Another feature is that it has an affinity for tenacious adherence to metal surfaces, a quality which combined with exceptional film strength effectively prevents galling and seizing at bearing pressures in excess of the yield strength of most metals.

MOLYKOTE, Type Z, may be applied dry in the powder form or it may be mixed with solvents, oils or greases. It is finding increasing use as a wear-in lubricant for dies, gears and machinery. It excels as a thread lubricant, especially on threaded connections subjected to high temperatures for sustained periods. It effectively prevents freezing of threaded connections. Applications of MOLYKOTE have made it easy to do some really tough jobs of drawing and cold forming stainless steel and other materials.

Available from The Alpha Corporation, Greenwich, Connecticut.

HYDRAULIC HANDBOOK

E. F. Houghton & Co., manufacturer of industrial oils, chemicals and leathers, has issued a revised edition of the *Houghton Handbook on Hydraulic Oils.*

The new edition has one outstanding addition — a chapter on flushing of hydraulic systems. It discusses the different procedures used in flushing of hydraulic systems which have become contaminated, and recommends an improved procedure.

Other chapters in the revised book deal with hydraulic oil specifications, the importance of properties not in specifications, additive treatment of hydraulic oils, trouble shooting, the qualities necessary in a good hydraulic oil and a discussion of Hydro-drive hydraulic oils.

The handbook, attractively bound in orange and blue, is available upon request to E. F. Houghton & Co., 303 W. Lehigh Ave., Philadelphia 33, Pennsylvania.

(*Continued on page* 246)

LUBRICATION ON PARADE!

NEW MAINTENANCE SHOW

The first Plant Maintenance Show, and exposition devoted exclusively to cost reduction through improved installation, operation and maintenance of equipment and services in factories, warehouses and other plants, will be held in the Auditorium, Cleveland, O., January 16-19, inclusive, it was announced recently by Clapp & Poliak, Inc., of New York, exposition management. Concurrently with the show, a four-day conference on plant maintenance methods will be held with L. C. Morrow, editor, *Factory Management & Maintenance,* as general chairman.

Exhibits and conference subjects will cover air conditioning, heating and ventilating; buildings, building materials and services; electrical equipment; employe relations, training and safety; instruments and meters; lubricants and lubricating equipment; maintenance tools and supplies; management consultation services; materials handling methods and equipment; mechanical rubber goods, including belts, hose and packing; paints, painting equipment and product finishes; power generation and distribution; power transmission, and welding and gas cutting.

The show and conference will emphasize those aspects of plant operation not concerned with direct production but whose share in total costs is a considerable factor. Those invited will be principally vice-presidents in charge of operations and manufacturing, production managers, plant managers, maintenance engineers, superintendents, chief engineers, and others who are concerned with operation and maintenance of equipment and services needed for efficient, low-cost manufacturing.

"The Plant Maintenance Show and the Conference are a response to the urgent demands of management in all industries for thorough, systematic programs of cost reduction," said Saul Poliak, vice-president

"Plant maintenance, although integral with production, has generally received less attention than other, more dramatic—but certainly no more significant—phases of production. Potentialities for cost reduction through better equipment, methods and processes for plant maintenance are immense. The show and conference will make a twofold contribution. First, through showing new machines, equipment, materials and services that can bring maintenance functions to maximum effectiveness. Second, through an exchange of ideas and information about advanced techniques of maintenance."

A.S.L.E. PARTICIPATION

The American Society of Lubrication Engineers will be represented at the conference with a panel discussion on improved lubrication practice. The speakers will be outstanding and authoritative men in their field.

Our exhibit at the show will present a graphic and dramatic picture of the progress of lubrication, from the invention of the wheel to the most modern principles of "metals in motion."

There will be no charge for either exhibit or conference attendance. Advance registration cards may be obtained from Clapp & Poliak, Inc., 350 Fifth Avenue, New York 1, N. Y.

HYDRAULIC CONFERENCE

The Fifth Annual Meeting of the National Conference on Industrial Hydraulics, the only technical meeting where engineers who work in industrial hydraulics can discuss all their mutual problems, will be held October 26-27, 1949, at the Sheraton Hotel, Chicago.

The field of industrial hydraulics includes a part of practically every field of engineering and industry, and thus far the Conference is the only common meeting ground. The Conference is intended to foster an interchange of ideas, methods of approach and new techniques in the field by the presentation of technical papers.

Otto J. Maha, vice-president of the Hannifin Corp., Chicago, is director of the Conference. Sidney F. Musselman is conference secretary and assistant chairman of the applied mechanics research department of Armour Research Foundation of Illinois Institute of Technology, which, with the graduate school of Illinois Tech and the local sections of seven technical societies, sponsors the non-profit, non-commercial conference.

The seven societies which cooperate in sponsoring the meetings are the Western Society of Engineers, American Society of Lubrication Engineers and the local sections of the American Society of Civil Engineers, American Society of Mechanical Engineers, Society of Automotive Engineers, American Institute of Chemical Engineers, and the Institute of Aeronautical Sciences.

The two-day program will present 15 technical papers in five sessions. The sessions include: RECENT DEVELOPMENTS, HYDRAULIC EQUIPMENT COMPONENTS, DESIGN CONSIDERATIONS, HYDRAULIC EQUIPMENT STANDARDS, HYDRAULIC CIRCUIT PERFORMANCE.

Requests for copies of the program, purchase of Proceedings, and correspondence relative to the general activities of the Conference or to registration for the Conference should be addressed to Mr. S. F. Musselman, Conference Secretary, N.C.I.H., Armour Research Foundation, Technology Center, Chicago 16, Illinois.

MAAG SPEAKS

The second anti-friction bearing symposium was held September 6-9 at Canton, Ohio, under the auspices of the Timken Roller Bearing Company. This conference was attended by a group of the leading educators in Mechanical Engineering and Machine Design. Lectures were given by a group of distinguished specialists covering all phases of anti-friction bearing manufacture and operation. Oscar L. Maag, a recent past president of A.S.L.E., lectured on various aspects of lubrication and led the discussions on this phase of bearing operation.

CHEK-CHART SALUTE

Chek-Chart's 20th Anniversary Celebration; a salute to the automotive industry, will be held at the Hotel Statler, Detroit, October 20. This falls within Oil Progress week, October 16-22 in which the Petroleum Industry marks its 90th year. Members of the A.P.I. Lubrication Committee and the National Oil Industry Information Committee will be present.

CURRENT LITERATURE

Journal of Applied Physics, Vol. 20, No. 4, April, 1949

THERMOELECTRIC EXPERIMENTS WITH EXTREME-PRESSURE LUBRICANTS
Schnurmann, R.—pp. 376-383

Reference is made to the conjectural nature of our present views on the mechanism of extreme-pressure lubrication. A differentiation is made between sliding and tearing so that sliding is defined as the relative motion of two solid bodies in frictional contact under the action of a tangential force with the frictional force increasing in direct proportion with the normal load. On the other hand, when the surfaces of the friction elements suffer microscopic damage during relative motion, the frictional force increases at a more than proportionate rate with the normal load and a case of tearing is established. The general requirement made of a lubricant is the reduction of the frictional force. The requirement made of an extreme-pressure lubricant is that it should reduce the frictional force at a sufficiently high value of the normal load for a pressure of at least 1.5×10^4 kg-wt./cm^2 to apply throughout the run, and in addition, the extreme-pressure lubricant should prevent tearing or scoring of the surfaces at these comparatively high bearing pressures. Evidence was found that extreme-pressure addition agents reduced the frictional force and extended the region of normal loads over which the frictional force was a direct proportion of the normal load. Various types of extreme-pressure addition agents gave results which suggested the existence of an optimum dope concentration. In general, over the region of normal loads over which Amonthns' law of the direct proportion between the frictional force and the normal load applied, lubricants which reduced the thermal electromotive forces generated at the rubbing contact also reduced the wear diameter. On the other hand, when tearing took place no such parallelism between friction and wear became apparent.

Journal of the Institute of Petroleum, Vol. 35, No. 303, March, 1949

DETERGENCY OF CARBON BLACK IN HYDROCARBON SOLUTION. Part I
Baddeley, A. R. W., Nissan, A. H., and Garner, F. H.—pp. 141-170

The ability to retain carbon in suspensions of lubricating oil is commonly described as detergency, although the problem is one of both cleansing and dispersion. Suspensions of carbon black (Micronex) in toluene solutions of calcium naphthenate were studied in order to determine the degree of dispersion and the adsorption of calcium naphthenate on the carbon black. The degree of dispersion was determined by measuring the rates of settling of the suspensions over a period of time of forty-eight hours. The adsorption of calcium naphthenate on the carbon black was determined by measuring the concentration of calcium naphthenate in the clear solution from which the carbon black had settled out; force-area curves obtained on the Langmuir trough were used to determine the concentration. The degree of dispersion as measured above and the adsorption of calcium naphthenate on the carbon black approached a maximum at about 0.188% calcium-naphthenate concentration in toluene. On pp. 160-170 the reader will find the discussion of the paper.

REACTION OF SULFUR, HYDROGEN SULFIDE, AND MERCAPTANS WITH UNSATURATED HYDROCARBONS. LITERATURE SURVEY
Hoffert, W. H., and Wendtner, K.—pp. 171-192

Presenting one hundred and thirteen references the authors survey the literature on the reactions of sulfur, hydrogen sulfide, and mercaptans with unsaturated hydrocarbons which shows that sulfur commences to react with some unsaturated hydrocarbons at temperatures below 100°C. The primary reaction is the formation of polysulfides. These sulfur compounds break up at higher temperatures to give alkyl mercaptans, dialkyl sulfides, and hydrogen sulfide, which therefore are secondary and not primary products of the reaction as assumed hitherto. Thiophen derivatives are also formed from some unsaturated hydrocarbons at temperatures of 150° to 200°C.

REVERSIBILITY OF THE PROPAGATION REACTION IN POLYMERIZATION PROCESSES AND ITS MANIFESTATION IN THE PHENOMENON OF "CEILING TEMPERATURE"
Dainton, F. S., and Ivin, K. J.—p. 100A
(Nature, 1948, Vol. 162, pp. 705-707)

In studying the copolymerization of olefins and sulfur dioxide, theoretical considerations suggest that, as the temperature increases, the reaction rate should first rise exponentially then pass through a maximum, and fall to zero at a ceiling temperature. The kinetic and thermodynamic arguments which lead to this conclusion are outlined. Experimental data in its support are given for 1-butene, 2-butene, and propylene.

FUEL EVALUATION ON THE BASIS OF VAPOR LOCK TENDENCY
Schneider, K. W., and Grunwald, A.—p. 103A
Erdol u. Kohle, Vol. 1, 1948, pp. 116-120

Factors influencing the cut-off point at which the vapor lock in a fuel system interrupts fuel supply are discussed. A laboratory apparatus is described whereby the influence of temperatures and pressure on the V/L ratios (V-vapor volume, L-liquid volume delivered by the fuel-supply system) of fuels in the range 1 to 20 can be determined. The results obtained with this apparatus enable curves to be constructed whence the behavior of a fuel in various types of engines and under various operating conditions can be predicted.

IMPROVED VISCOMETER
Jacobs, H. R.—p. 103A
(Rev. Sci. Instrum., Vol. 19, 1948, p. 816)

The viscometer is described and its advantages are indicated. These advantages are: (1) hydrostatic effects of density are eliminated; (2) surface tension effects are minimized; and (3) rates of flow can be measured over a wide range of pressures.

KNOCK PREVENTER
Anon.—p. 106A
(Amer. Automobile, Vol. 25, No. 11, Nov., 1948, p. 25)

Based on the system established during the war whereby injection of 50/50 methanol-water into fuel allowed aircraft to have much increased power for short periods, the instrument known as the Vitameter has been developed for use by motor cars. This accessory is actuated by drop in inlet-manifold vacuum on open throttle and feeds measured quantities of Vitol (85% methanol, 15% water containing three cubic centimeters of tetraethyllead per gallon and a soluble oil). This injection serves to raise the rating of the entire fuel mixture by 10-20 octane number, thus allowing modern engines of relatively high-compression ratio to use low-grade gasolines. An added advantage is that, using Vitol, engine deposits are soft and easily removable.

COLD STARTING
Ridley, C.—p. 106A
(Auto. Engr., Vol. 39, 1948, p. 21)

It has been found that ease of engine starting under cold conditions can be greatly increased by continuously forcing a stream of a relatively wide-cut mixture of several volatile hydrocarbons of differing boiling points into the intake manifold, quite independently of the air stream. The hydrocarbons to be contained by the mixture which is most suitable for this purpose, are indicated.

"KNOCK" IN INTERNAL COMBUSTION ENGINES
Downs, D., Walsh, A. D., and Wheeler, R. W. —p. 106A
(Nature, Vol. 162, 1948, p. 893)

Gas samples have been extracted from the cylinder of a Richardo E6 variable-compression engine operating at 1500 revolutions per minute. Their aldehyde content and total organic and hydrogen peroxides have been measured. Using technical iso-octane as fuel, it is found that as the compression ratio is increased, "knock" develops and the peroxides concentration increases, giving also a much higher peak value near the point of flame arrival. The aldehyde concentration of the gas is parallel with that of the peroxides, but the peak concentration of the former is some thirty times greater than that of the peroxides. The effect of tetraethyllead addition is to lower the concentration of both peroxides and aldehydes.

KEEPING THE WHEELS TURNING; THE VITAL PART PLAYED BY LUBRICATING OIL ADDITIVES
Hope, A. E.—p. 107A
(Scientific Lubrication, Vol. 1, Oct., 1948, pp. 11-13, 20)

The development of motor lubricating oils is described, and the properties of solvent-refined lubricants indicated. The special functions of engine testing and of chemical additives are outlined and the respective fields of regular, premium and heavy-duty oils differentiated.

APPLICATION OF COLLOIDAL GRAPHITE TO SOME LUBRICATION PROBLEMS
Stuart, A. H.—p. 107A
(Scientific Lubrication, Vol. 1, Oct., 1948, pp. 15-17, 24)

Phenomena of pressure distribution, film rupture, and seizure under conventional lubricating conditions are discussed, and the special properties of graphite as a lubricant are enumerated. Applications to fabric bearings, running-in new machinery, upper-cylinder lubrication, and metal drawing are described.

FUTURE LUBRICANTS
Vineall, G. J. C.—p. 108A
(Scientific Lubrication, Vol. 1, Oct., 1948, pp. 7-10)

The historical development of lubricants up to solvent-refined, extreme pressure, and hypoid oils is outlined, and the functions of engine tests and additives are discussed. Applications of synthetic oils

are indicated, and Ucon, Prestone, and Plexol oils are described together with silicones. The theoretical basis of recent developments and the function of metallic soaps are discussed.

LUBRICATION OF LAND AND MARINE OIL ENGINES
Anon.—p. 108A
(Scientific Lubrication, Vol. 1, Oct., 1948, pp. 3-6)

The larger type of vertical multi-cylinder totally enclosed dry-sump engine is considered as typical installation. The effects of modern design and operating conditions on lubrication requirements are stressed, and pressure-circulating systems are described. The system employed for crankshaft lubrication is outlined.

AIR AND OIL FILTRATION
Anon.—p. 112A
(Auto. Engr., Vol. 39, 1948, p. 12)

This note contains a table of the results of wear tests on an engine equipped with a Vokes air filter and a partial-flow oil filter. From four trials it was found that, although the partial-flow filter removes considerably more solid matter from the lubricating oil, it does not protect the bearings to the same extent as the full-flow filter.

Lubrication, Vol. 35, No. 5, May, 1949

MACHINE TOOL LUBRICATION ON THE PRODUCTION LINE
Anon.—pp. 49-60

The theme of this article presents a means for maintaining quality lubrication, yet reducing costs of maintenance and the total number of lubricants used in a given production shop to a minimum. The formulation of a simplified lubrication plan must of necessity include a thorough understanding of the operating characteristics of machine tools and an intimate knowledge of lubricant recommendations issued by individual machine tool builders. Such a plan could not be effective without the availability of recently developed lubricants, particularly of the additive type. These lubricants can be used over a much wider range of operating conditions than was possible with lubricants available not too long ago.

Mechanical Engineering, Vol. 71, No. 4, April, 1949

THE LOAD-CARRYING ABILITY OF HYDRODYNAMIC OIL FILMS
Roach, A. E.—pp. 293-296, 326

The author presents a very brief survey of the progress and present state of the hydrodynamic theory of lubrication, particularly as it relates to the load-carrying capacity of journal bearings and reviews some current problems. Four criteria for load-carrying capacity, effect of variation of viscosity, effects of surface conditions, and load conditions and speed conditions are discussed. Forty-three references are given. The paper was presented at the Annual Meeting, New York, N. Y., November 28-December 3, 1948, of The American Society of Mechanical Engineers.

APPLICATION OF STATISTICS TO THE SCIENCE OF MANAGEMENT
Juran, J. M.—pp. 321-324

The paper was presented at the Annual Meeting, New York, N. Y., November 28-December 3, 1948, of The American Society of Mechanical Engineers. Pointing out that the science of statistics plays an important role in the use of the scientific method, and stressing the importance of applying scientific method (and therefore statistics) to the industrial management, the author exemplifies the application of the method advocated to the management program of a company in the textile industry.

A CORRELATION OF VISCOSITY, TEMPERATURE, AND PRESSURE
Sanderson, R. T.—p. 349

Stating that attempts to correlate absolute viscosity, temperature, and pressure quantitatively in a form conveniently applicable to lubricating oils have not been entirely satisfactory, and that kinematic viscosity appears to conform to a simple relationship contributing to a fundamental understanding of viscosity and promising valuable practical applicability, the author suggests a method for correlation of viscosity, temperature and pressure which, at least approximately, is correct.

National Petroleum Association, Saturday News Roundup, May 7, 1949

PENNSYLVANIA GRADE ASSOCIATION ASKS MILITARY TO STOCKPILE LUBES
Anon.—pp. 1-2

Pennsylvania Grade Crude Oil Association has requested the Armed Services Petroleum Board to stockpile lubricants. This would be advantageous both from a price standpoint and from the standpoint of conserving an important natural resource. This is based on the recent drop in prices of both crude and products from the Pennsylvania field. It is unprofitable to operate many of the producing properties in the field. The Association pointed out that over two-thirds of the current production of Pennsylvania grade oil is produced by means of secondary recovery operations.

National Petroleum News, Vol. 41, No. 21, May 25, 1949

GM (GENERAL MOTORS) TELLS HOW OIL COMPANIES CAN MAKE, MARKET AUTOMATIC TRANSMISSION FLUID
Ridder, H.—pp. 13, 49

The author reports that API Lubrication Committee on May 23, 1949, heard R. H. Wolf, of General Motors Research, explain procedure by which oil companies may manufacture or market fluid for GM automatic transmissions through service stations. A bulletin is issued by GM where, under the heading "Qualification Procedure," eleven points are listed which explain how to obtain permission to market the said fluid. Between now and January 1, 1950, when new Type A fluid for transmissions will be made available to oil marketers, marketers can apply to GM for permission to obtain from two present suppliers current fluid and rebrand it for sale under their own name. The committee approved a motion made by B. G. Symon, Shell Oil, that panel be set up to study problem of reclaimed oil.

OIL CHANGE PRACTICE, ITS EFFECT ON ENGINE CONDITION
Howe, W. A., Drake, J. F., Jr., and Donaldson, H. H.—pp. 33-34

The partial text of paper read before API Lubrication Committee meeting May 22-24, 1949, at Colorado Springs, Colorado, is presented. The author reviews the lubricating oil drain recommendations and the advantages of a shorter drain intervals. Average conditions, oil consumption, increased contamination of oil with metals, abrasives and road dirt the longer it is used in the engine before being changed, and savings resulting from shorter drain intervals are discussed.

Oil and Gas Journal, Vol. 47, No. 51, April 21, 1949, Natural Gasoline Number

SHELL OIL'S NEW FIELD LABORATORY EXPEDITES DEVELOPMENT PROGRAM
Stormont, D. H.—pp. 192, 193, 195

A growing need for prompt sample analysis and laboratory work tailored to the requirements of the area has prompted the establishment early this year of the Shell Oil Co., Inc., field laboratory at Odessa, Texas.

DOMESTIC AND FOREIGN PERIODICALS IN THE FIELD OF PETROLEUM CHEMISTRY. A STATISTICAL ANALYSIS
Boig, F. S., and Loftman, K. A.—pp. 199, 200, 204, 207, 208

This is an interesting article on the journals of interest to the petroleum industry. It attempts to go into the following questions: 1. What are the important domestic and foreign journals in this field? 2. What is their relative importance? 3. What countries lead in publications in this field? 4. What is the necessity for the knowledge of a foreign language and what language? 5. Is there any visible effect of the last war on publications and research in this field? 6. If a large petroleum company decided to subscribe to foreign petroleum journals for its literary or reference files, which periodicals seem to be the most important? The limitation of this article is that it is based exclusively on articles abstracted in the Chemical Abstracts, which does not cover certain periodicals of interest to the petroleum industry well. This work is one of a series of research investigations dealing with the frequency and source of publication of technical articles. A similar study on paints and varnish was published by the "Official Digest, Federation of Paint and Varnish Production Clubs": an article in the field of analytical chemistry will appear in the March issue of Chemist-Analyst, and one on dyes and textiles chemistry has been accepted and will soon appear in the Textile Research Journal.

VISCOSITY OF RESERVOIR OILS
Calhoun, J. C., Jr.—p. 245

One of the basic ideas in the analysis of reservoir behavior is that the numerical values which are assigned to physical properties of reservoir fluids are dependent upon the pressure and temperature of the reservoir. In previous articles in this series the effect of these variables upon gas solubility and formation volume factor have been discussed. In the present discussion the variation of the viscosity of reservoir oil is considered.

Vol. 47, No. 52, April 28, 1949

FUTURE OF MEXICAN ECONOMY DEPENDS ON PROMPT DEVELOPMENT OF OIL RESOURCES
Irizzary, O. B.—pp. 70-72

LUBE-OIL PRODUCERS CONCERNED OVER MARKET OUTLOOK AND LARGE CAPACITY
Weber, G.—pp. 78-79, 183

The sudden switch from a firm lube position to one of current oversupply has surprised many refiners. While a difference of opinion still exists as to the ultimate result of present trends, many observers believe that a major relocation of the lubricating-oil industry is definitely under way. Technology plays a major part in this appraisal. Modern methods of vacuum distillation, extractive refining, and dewaxing have widened the base of crude types suitable as lubricant sources. Moreover, modern manufacturing units

(*Continued on page* 242)

NEWS OF THE SECTIONS

DETROIT, 1950 A.S.L.E. CONVENTION

The Detroit Section of the ASLE is making excellent progress in its role as host for our next annual convention. Mr. Edwin D. Harkins, Chairman of the Section, and Mr. James W. Hopkinson, Chairman of the Convention Committee, along with the other officials and members, are to be congratulated on their advance activities to date. The Convention Committee and Section Officers met in July at the Hotel Statler, Detroit, with W. F. Leonard, National Secretary, at which time plans were outlined. There will be a suitable headquarters room for the delegates' wives and at all times, some of the Detroit Section members' wives will be in attendance. Tentative arrangements have been made for a Fashion Show and a visit to an automobile manufacturing plant. We strongly urge all members and interested guests to watch the December issue of LUBRICATION ENGINEERING for further announcements from Detroit. The success of any convention depends to a great extent on the support of the local group of the city in which the meeting takes place. The Detroit Section is full cognizant of this fact, and will serve "over and beyond the call of duty," thus sponsoring the growth—not only of their local membership—but of the National as well.

INDIANAPOLIS

A schedule of meeting dates for the Indianapolis Section is listed below:

Monday — September 26th
Monday — October 31st
Monday — November 28th
No December meeting scheduled because of conflict with Christmas holidays.
Monday — January 30th
Monday — February 27th
Monday — March 27th
Monday — April 24th
Monday — May 22nd

Information as to the speakers is not yet available.

SECTION SECRETARIES

With the close of the summer season, all the Sections will again be resuming their monthly meetings. May we urge all Section Secretaries to send to the National office, plans for the winter season. Material must be received no later than the 10th of the month prior to publication.

BUFFALO

New Officers for the 1949-50 season were installed at the September meeting of this Section. (See Section Officers' page) James E. Farry, retiring Chairman, presented the new Officers who were elected by letter ballot during the summer. Much enthusiasm was evidenced in support of a new Section which is being planned in Syracuse, New York. The turn-out for the opening fall meeting was a good omen for an active and progressive year ahead. OCTOBER MEETING— Dr. M. E. Merchant of the Cincinnati Milling Machine Co. will deliver a paper on Cutting Fluids. NOVEMBER MEETING — The Council of Technical Societies of the Niagra Frontier stated that the Council is planning a joint meeting of all Societies represented, for November.

MILWAUKEE

The first meeting of the 1949-50 season had as its speaker Mr. Ralph Matthews, Executive Vice-President of the Battenfeld Grease & Oil Corp. His subject, "Multi-Purpose Greases," was well received. The interest manifested at the opening session was indicative of a successful year ahead for the Section.

FORT WAYNE

The September meeting of this Section, opening the fall season, was an auspicious occasion. The speaker was our A.S.L.E. President, Mr. Charles L. Pope, of the Eastman Kodak Company. Mr. Pope's talks always assure an interesting evening evocative of spirited discussion.

DETROIT

A bulletin from Detroit and we quote—

"The month of August is not particularly noted for membership activity with most people on vacation, but our Membership Committee has been quite active, and congratulations are due the following new members: Robert Moore, Albert Angelus, Russell Hopkinson, George Hynes and Hugh Wagner."

The annual stag outing was held on September 24th at the Hillcrest Golf and Country Club in Mount Clemens, Michigan. The stag day party has become a "must" for members and friends of the Detroit group. The large attendance either played golf, softball, or took part in the horseshoe contest, and the general spirit of good fellowship was evident everywhere, and not just confined to the 19th hole. The Papers Committee, under the able Chairmanship of Mr. P. W. Uhl, arranged a spectacular line-up for the fall sessions, starting with a primary lubricants meeting in September, and progressively adding talks on General Plant Lubrication, Hydraulic Oils, Lubrication of Plain and Anti-Friction Bearings, Internal Combustion Engine Lubrication, Cutting Fluids, Drawing Compounds and Cleaning Methods, and a wind-up session of a discussion of talks given at the previous meetings.

CHICAGO vs DAYTON

The NATIONAL OFFICE has just received notice that the membership contest between these two Sections is again on for the 1949-50 season. Dayton sends Chicago fair warning that the gavel which they won at the close of the 1948-49 round is much appreciated but they do have need of other furnishings and equipment. We are in the neutral corner. Good luck.

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CURRENT LITERATURE

(*Continued from page* 240)

are characterized by a large volume throughout. Installed logically in refineries of major size, they represent huge investments which must be returned by capacity operations. Their production will greatly reduce the outside purchases of base stocks which formerly supplied many companies with blending stocks for their branded oils. To move toward greater self-sufficiency on the part of former buyers in the open market stems in part from the dislocation of supply both during the war and during the seller's market immediately following.

Vol. 47, No. 53, May 5, 1949

32—COMPANY REPORT SHOWS NEW RECORDS AGAIN SET IN 1948
Sansone, F. J.—pp. 48-50

The principal financial features of the petroleum industry's operations in 1948 as revealed by the annual reports of a group of 32 oil companies may be summarized as follows: 1. New high records were established in practically every financial as well as operating phase of the business. 2. An increase of 35 per cent in the price index for crude oil and petroleum products and a gain of 4 per cent in total petroleum demand combined to produce a rise of 35 per cent in operating revenues in 1948 over 1947. 3. Net income after all charges in 1948 increased 54 per cent. 4. Cash dividend disbursements increased 14 per cent but represented only 26 per cent of net earnings or a smaller portion than in any other year. 5. Despite record income and retention of 74 per cent of net earnings, funds available for operations were again inadequate.

FLASH POINT OF BLENDS
Nelson, W. L.—p. 101

How can we compute the flash point of blends of lubricating oil? The only source of published experimental information is the paper "Prediction of the Flash Point of Blends of Lubricating Oils," Ind. Eng. Chem., Vol. 19, 1927, p. 259. In this a formula is suggested.

Vol. 48, No. 1, May 12, 1949

HALF OF CARS NOW REQUIRE PREMIUM GASOLINE, A. P. I. MARKETING MEN TOLD. (St. Louis meeting)
Anon.—p. 55

About half of all cars on the road require premium gasoline for knock-free performance, Julian J. Frey, general sales manager of Ethyl Corporation, told the Marketing Division of the American Petroleum Institute at its meeting May 10. Citing independent surveys by Socony-Vacuum Oil Company, Inc., Shell Oil Company, Inc., and Ethyl Corporation, Frey declared the data accumulated "indicate that only 30 to 50 per cent of the present car population will be free of knock on the current regular grades of motor fuels."

MAJOR EXPANSION PROGRAM UNDER WAY WITHIN BRITAIN'S CHEMICALS INDUSTRY
Duff, D. M.—pp. 62-63

Development of a petrochemical industry in Britain complements the expansion now being made in the primary crude-oil refining industry itself. New plant in both the chemical and petroleum-refining fields are being encouraged by the government. Last year Britain exported about $335,000,000 in chemicals, drugs, and dyes, compared to about $270,000,000 in 1947 and about $90,000,000 in 1938.

DETROIT

The dynamic industrial city, is the A.S.L.E. Convention City for 1950

The dates are April 10-11-12 at the Hotel Statler

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Register Sunday and avoid the Monday morning rush.

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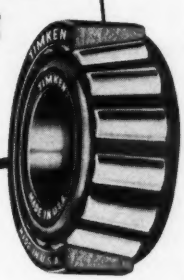
HOW TO SIMPLIFY LUBRICATION IN ALL KINDS OF MACHINERY



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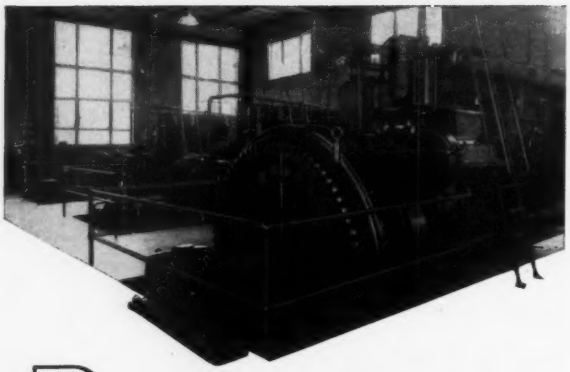

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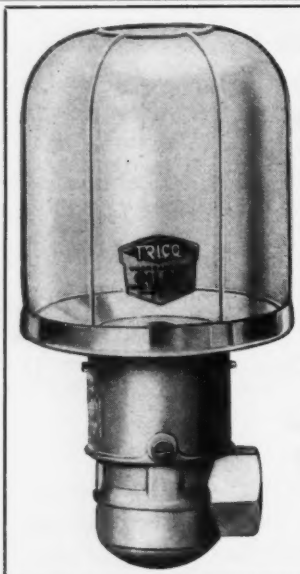

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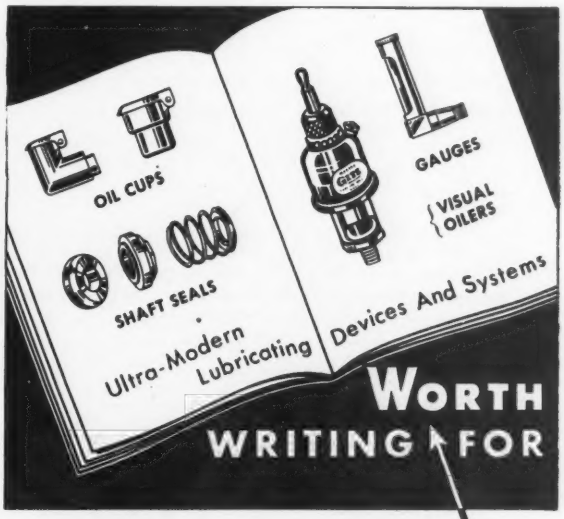

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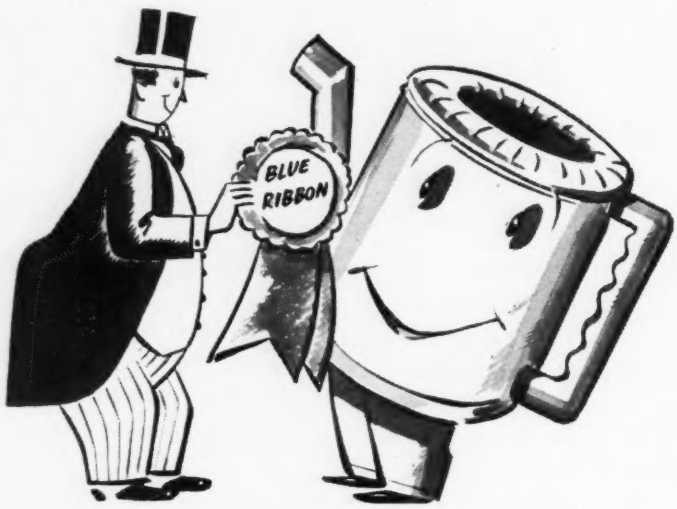

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New Products and Equipment

(*Continued from page* 237)

LUBRICATION AT 400 F.

Lubrication of industrial units in the temperature range above 400 F. is the subject of a bulletin, "dag" Colloidal Graphite for Industry, offered free by Acheson Colloids Corporation, Port Huron, Michigan.

At high temperatures, where ordinary lubricants begin to fail, "dag" colloidal graphite dispersions retain their lubricating properties and thereby reduce maintenance troubles. Also excellent for general lubrication, parting, impregnating, opaquing, and use as an electrical conductor.

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Automatic lubrication for millions of pressure lubricated plug valves is now possible without use of mechanical accessories or force-feed mechanisms. The secret is in a new "energizable" valve lubricant which provides fully automatic action to maintain anti-leak sealing efficiency and to reduce maintenance costs.

No lubricant heretofore known has the characteristics for storing within itself energy to be used in distributing its lubricating qualities throughout a valve system over a prolonged period of time. This new compressible lubricant, christened Hypermatic is said to allow automatic lubrication to be established in valves of the Nordstrom type. The manufacturer predicts this amazing substance, which is available in bulk or grease stick form, to cut lubrication costs from 20 to as high as 90 per cent. In many instances valves may be serviced only 1/10 as often.

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OILS AND LUBRICANTS PATENT ABSTRACTS

Ethylene Polysulfide Polymer Lubricant — Patent #2,474,859—Blyde A. Perkins, assignor to Rockwell Mfg. Co.

The process of preparing a synthetic ethylene polysulfide polymer of desired viscosity for use as a plug valve lubricant comprising reacting an aqueous sodium polysulfide solution having a sodium polysulfide ratio of $Na_2S_{3.5}$ to $Na_2S_{4.0}$ with ethylene dichloride to produce a polymeric ethylene polysulphide material and heating said material at in the range of 380°F. to 400°F. for between one and one-half to two hours to convert it into a lubricant of said desired viscosity.

Production of Viscous Oil by Copolymerization of Aromatic Olefin with Monolefin — Patent #2,474,881—David W. Young and William H. Smyers, Assignors to Standard Oil Development Co.

A process for producing synthetic hydrocarbon oils which comprises copolymerizing a mixture of 70 to 80% by weight of isobutylene and 30 to 20% by weight of styrene by refluxing at a temperature of 10° to 180°C. in the presence of a boron fluoride catalyst.

Lubricants — Patent #2,475,246 — Louis A. Mikeska, assignor to Standard Oil Development Co.

A lubricant consisting essentially of a major proportion of a mineral lubricating oil and 0.1 to 10% by weight of an oil soluble polymer of a monomeric material having the general formula $CH_2CXCOSR$ in which X is a radical selected from the class consisting of hydrogen and a lower alkyl group and R is an alkyl group having a carbon content in the range of 5 carbon atoms to the atomic carbon content of paraffin wax, said polymer having a molecular weight in the range of 1000 to 50,000.

Decolorization of Oil with Magnesium Silicate —Patent #2,475,328—William A. LaLande, Jr., assignor to Attapulgus Clay Co.

A method of decolorizing oil, which comprises contacting said oil, at a temperature between 200°F. and 500°F., with a water-insoluble magnesium silicate activated by treatment with an ammonium compound yielding NH_4+ions in aqueous solution at a temperature between 150°F. and the boiling point of the solution, and thereafter washed with water to remove the soluble salts, and dried.

Lubricants — Patent #2,475,410 — Herschel G. Smith, Troy L. Cantrell, and John G. Peters, assignors to Gulf Oil Corp.

As an improved compound lubricant, a uniform blend of a mineral oil lubricating base with a small amount, sufficient to retard rust and corrosion, of a product obtained by reacting a mixture of sulfurized unsaturated fatty acid containing over 8 carbon atoms and an aliphatic alcohol containing over 8 carbon atoms with a member of the group consisting of maleic acid and maleic anhydride, then reacting the product thus produced with a member of the group consisting of olefin oxides containing less than about 8 carbon atoms and glycols corresponding thereto, both of said reactions being carried out at a temperature between about room temperature and about 240°F. and then neutralizing the reaction product with an aliphatic primary amine containing between about 8 and 26 carbon atoms.

Lubricating Grease Composition — Patent #2,475,589 — Arnold A. Bondi, assignor to Shell Development Co.

A mechanically stable, thermally reversible non-bleeding lithium soap grease comprising a major amount of a mineral lubricating oil, from about 5% to 25% of a lithium soap, of a mixture comprising from about 10% to about 90% of 12-hydroxy stearic acid and from about 90% to about 10% of a hydrogenated fish oil fatty acid, and from 0.1 to 1% of a polyalkylene glycol having an average molecular weight of between about 400 to about 7000.

Oil Base Drilling Fluid and Mixing Oil for the Same—Patent #2,475,713—George Miller, assignor to Oil Base, Inc.

An oil base drilling fluid of the type employing blown asphalt as a plastering agent, a weight material and a petroleum mixing oil in proportions to produce a fluid oil in which the mixing oil is within the gravity of 29° to 40° API, flash 130° to 180°F., initial boiling point 320° to 420°F., final boiling point 500° to 650°F., aniline point 90° to 150°F., surface tension less than 24 dynes per cm. at 20°C. and an acid number below 0.2, the drilling fluid being further characterized by the addition of sufficient napthenic acid to bring the acid number between 0.2 and 1.5.

Lubricating Oil Compositions and Improvement Agents Therefor — Patent #2,475,727 — Herschel G. Smith and Troy L. Cantrell, assignors to Gulf Oil Corp.

An improved liquid lubricating oil composition comprising a mineral lubricating oil and an anhydrous, non-hygroscopic mixture of sperm oil alcohols and the calcium salts of sperm oil fatty acids, the fatty acids in the form of their calcium salts and the sperm oil alcohols existing in substantially the same relative proportions as the corresponding alcohols and acids exist in the unsaponified sperm oil, and being present in an amount sufficient to confer improved detergent, antioxidant and oiliness properties to the composition.

Printed copies of patents are available from the Patent Office at twenty-five cents each. Address the Commissioner of Patents, Washington, D. C., for copies and for general information concerning patents.

From Official Gazette—Vol. 624, Nos. 1, 2, 3, 4. Vol. 625, Nos. 1, 2, 3, 4, 5.

Prepared by ANN BURCHICK

Lubricants — Patent #2,475,970 — Eugene Lieber and Raymond M. Dean, assignors to Standard Oil Development Co.

A lubricating oil composition consisting essentially of a major proportion of waxy mineral lubricating oil containing dissolved therein a pour depressing amount of a Friedel-Crafts condensation product of from 5 to 10 mols of diisobutylene and 1 mol of an initial Friedel-Crafts condensate having a molecular weight of at least 1000 and being substantially non-volatile up to 600°F. obtained by condensing from 2 to 5 mols of a chlorinated paraffin wax containing from 10 to 20% chlorine and 1 mol of napthalene at a temperature ranging from room temperature to about 300°F., said recondensation of said initial condensate an diisobutylene taking place at a temperature ranging from 100 to 200°F.

Synthetic Lubricants by the Catalytic Treatment of Unsaturated Ketones—Patent #2,476,157 — Ferdinand P. Otto, assignor to Socony-Vacuum Oil Co.

The product formed by treating an unsaturated ketone having ten to thirty carbon atoms in each alkene radical with a catalyst selected from the group consisting of synthetic silica-alumina cracking catalyst and activated montmorillonite clay, at a temperature falling within the range varying between about 500°F. and about 650°F. and for a period of time varying between about three hours and about seven hours.

Mineral Oil Composition — Patent #2,476,696 — Solomon Caplan and Peter L. Rosamilia, assignors to The Harvel Corp.

A hydrocarbon ether of hydrogenated carbanol dissolved in a mineral hyrocarbon oil whose flash point is above 250°F. said ether being present in small amounts, sufficient to inhibit sludging of said oil.

Lubricating Composition — Patent #2,476,812 — John P. Buckmann and Loren L. Neff, assignors to Uaion Oil Co. of California.

A composition of matter adapted for addition to lubricating oil comprising an oil-soluble alkaline earth metal salt of the reaction product obtained by phosphosulfurizing a hydrocarbon lubricating oil fraction from which the most highly aromatic and the most highly paraffinic components have been removed, which hydrocarbon lubricating oil fraction is selected from those lubricating oil fractions obtained as solvent raffinate of a solvent extract and a solvent extract of a solvent raffinate, said solvent being a selective solvent of the type which selectivity dissolves the more aromatic hydrocarbons, and said selective solvent being chosen from the group consisting of the same solvent used in extract formation and raffinate formation under different conditions of treatment, and a different selective solvent used in extract formation and raffinate formation, said reaction product being obtained by reacting said lubricating oil fraction with between about 3% and 50% of a phosphorous sulfide at temperatures between about 200°F. and 600°F.

Lubricating Grease Compositions — Patent #2,477,311 — Arnold J. Morway, assignor to Standard Oil Development Co.

A lubricating grease composition comprising 60 to 90% mineral oil, 2 to 20% structural carbon black, selected from the group consisting of acetylene black and channel black of 10 to 50 millimicrons average diameter of abnormally high oil absorptive properties 1 to 30% petroleum resin obtained by propane extraction of deasphaltized petroleum residuums, said resin having a viscosity between 500 and 10,000 SSU at 210°F. and 0.1 to 5% of sulfur.

Turbine Oil — Patent #2,477,356 — Stephen J. Wayo, assignor to Sinclair Refining Co.

An improved lubricating composition adapted for use as a turbine oil comprising a petroleum oil having compounded therewith an alkylmercaptoacetic acid and an effective but relatively smaller quantity of an alkylsulfinylacetic acid, the alkyl group of each of the acids containing at least 12 carbon atoms, the acids being present in an amount sufficient to impart a corrosion-inhibiting property to the composition but not over 1% on the weight of the oil.

Mineral Lubricating Oil—Patent #2,477,913—Mintje van Loon, assignor to Shell Developing Company.

A lubricating oil comprising a mineral lubricating oil containing dissolved therein .1% to 1% of an oil-soluble composition produced by subjecting the zinc salt of di-isopropyl salicylic acid in the presence of a stable hydrocarbon oil to a temperature of about 300°C. and for a period of time of from ¾ to 1½ hours sufficient to effect the substantially complete redissolving of the precipitate formed by the decomposition of the salt in said stable hydrocarbon oil.

Treatment of Drying Oils—Patent #2,478,451 —Henry G. Berger and George S. Crandall and John F. Socolofsky, assignors to Socony-Vacuum Oil Co., Inc.

The process which comprises polymerizing a drying oil containing a minor proportion not in excess of about 0.2 per cent by weight of a polymerization catalyst of the Friedel-Crafts type at a temperature not substantially above about 300°F. to a body not substantially lower than Z-6 (Gardner-Holdt scale), arresting polymerization of said oil before completion of the reaction to gelation, while retaining said catalyst therein, by the addition thereto of a small quantity of an aromatic monocyclic amine.

Lubricant—Patent #2,478,694—Harris D. Hineline and David W. Young, assignors to Standard Oil Development Co.

The method of preparing a lubricant comprising the steps of polymerizing a polyolefin having from 6 to 14 carbons atoms per molecule to a molecular weight within the range between three thousand and several hundred thousand, cyclizing the polymer, chemically combining the cyclicized polymer with a phosphorus halide and dissolving .5 to 20% of the cyclicized chemically combined phosphorus compound containing polymer in a mineral base oil lubricating material.

Mineral Oil Composition—Patent #2,478,843—Raymond B. Seymour and Frederic L. Matthews, assignors to Monsanto Chemical Co.

A mineral oil having lubricating properties containing an oil soluble polymerization product of ar-cyclohexylstyrene in amount sufficient to increase the viscosity of said oil.

Rust Inhibiting Lubricants—Patent #2,479.424 —Lorne W. Sproule and Laurence F. King, assignors to Standard Oil Development Co.

A rust inhibiting lubricating composition consisting essentially of mineral lubricating oil containing 0.5 to 5% by weight, based on the total composition, of sorbitan mono-stearate and from 0.25 to 10% of an organic solubilizer and wetting agent for completely dissolving said stearate in said oil, said solubilizer being selected from the class which consists of at least one of the compounds sorbitan mono-oleate and pentaerythritol mono-oleate.

Lubricating Oil—Patent #2,479,513—Frederick P. Richter and Everett W. Fuller, assignors to Socony-Vacuum Oil Co., Inc.

A mineral lubricating oil containing a small proportion sufficient to improve the stability thereof, of an hydroxydibenzothiophene.

Drying Oils—Patent #2,479,857—Latimer D. Myers and James W. Ritz, assignors to Emery Industries, Inc.

The method of removing non-drying fatty acids from a fatty oil of the class consisting of drying and semi-drying oils, to improve the drying properties thereof, said method comprising heating the oil with polymer of fatty acids having, before their polymerization, the same number of carbon atoms per molecule as the non-drying fatty acids which are desired to be removed from the said oil, and then distilling from the said oil non-drying fatty acids which have been liberated by the said polymer acids.

Catalytic Oil Cracking—Patent #2,480,494—Eugene V. Mathy, assignor to Phillips Petroleum Co.

A method of cracking hydrocarbon oils of boiling point range higher than that of gasoline for the conversion of said oil into a high octane gasoline which comprises contacting the hydrocarbon oil in vapor form while at cracking temperature and pressure with an improved catalyst mass comprising as the essential catalyst components calcined bauxite supporting 0.1% to 5% by weight of tin oxide as a promoter.

Purification of Concentrated Oil Solutions of Polyvalent Metal Sulfonates—Patent #2,480,638—Gordon W. Duncan and John C. Zimmer, assignors to Standard Oil Development Co.

A method of removing inorganic salt impurities from an oil solution containing at least 20% of oil-soluble polyvalent metal sulfonates made by the double decomposition of alkali metal sulfonates and polyvalent metal compounds which comprises treating said oil solution containing the reaction product produced during said double decomposition reaction with an inorganic acidic material capable of forming water-insoluble salts with polyvalent metals and filtering to remove insoluble salts thus formed.

Method of Manufacturing Lubricating Grease — Patent #2,480,647 — George W. Gurd and Lorne W. Sproule, assignors to Standard Oil Development Co.

A method of preparing aluminum soap base grease which has a tendency to break down mechanically upon prolonged mechanical working during its transition from rubbery structure to gel structure, comprising heating aluminum soap with mineral lubricating oil at a grease-forming temperature, thereafter cooling said grease rapidly to a temperature approximating its transition temperature, accompanied by mechanical working, said cooling being accomplished by using an external cooling medium and thereafter mixing solid carbon dioxide into the grease with continued mechanical working to lower the grease temperature rapidly below said transition temperature at the latter part of the cooling operation.

Lubricating Oil Composition—Patent #2,480,664—John G. McNab and George M. McNulty and Thomas Cross, Jr., assignors to Standard Oil Development Co.

A mineral lubricating oil containing dissolved therein about 1% of a mixed alkaline earth metal salt of an alkylated phenol sulfide having an alkyl side chain of 4 to 8 carbon atoms attached to each benzene nucleus, the metal content of the salt consisting of 15% to 85% of calcium and 85% to 15% of a metal selected from the group consisting of barium and strontium, the percentages being molecular amounts, related to the total amount of metal required to neutralize the said alkylated phenol sulfide.

Transparent Metal Fabricating Lubricant — Patent #2,480,666—Arnold J. Morway and Hollis L. Leland, assignors to Standard Oil Development Co.

A light colored metal fabricating composition consisting essentially of a light colored mineral base lubricating oil containing 1 to 25%, based on the total composition, of a concentrate obtained by combining 10% light mineral oil with 5 to 20 parts by weight of sulfur, 40 to 85 parts of a C_1 to C_3 alkyl ester of abietic acid, and 2 to 15 parts of a metal sulfonate, heated to a temperature of 300°F. to 400°F. until the sulfur is substantially completely reacted with the ester.

Mineral Oil Lubricants—Patent #2,480,673—Orland M. Reiff and Harry J. Andress, Jr., assignors to Socony-Vacuum Oil Co., Inc.

The method of making a dithiophosphoric acid of an alkyl-substituted phenol which comprises: reacting an alkyl halide with phenol in the presence of a Friedel-Crafts catalyst to alkylate said phenol; washing the alkyl-substituted phenol product thus obtained with water to remove the Friedel-Crafts catalyst therefrom; reacting the catalyst-free, alkyl-substituted phenol with phosphorus pentasulfide to form a dithiophosphoric acid of the alkyl-substituted phenol; treating the dithiophosphoric acid of the alkyl-substituted phenol with at least about 1% of zinc and at least about 0.5% of water, based on the amount of alkyl-substituted phenol used, at a temperature of about 100°C.; and separating the zinc-treated dithiophosphoric acid of the alkyl-substituted phenol from the reaction mixture formed in the last-mentioned operation.

Milkweed Seed Oil—Patent #2,480,769—Cortes F. Reed, assignor to Federal Cartridge Corp.

Process of preparing oil from milkweed seed which comprises removing the fin from the seed, and then removing the oil from the seed while at a moisture content not substantially in excess of 8%.

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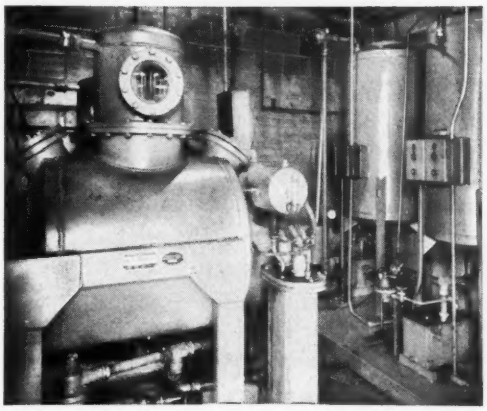
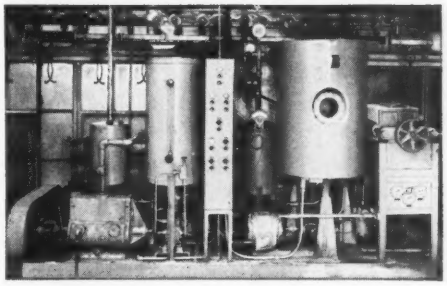


for **OILS** that *MUST BE* **EXTRA DRY**

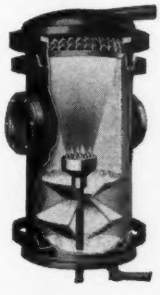
The Bowser degasifier shown above controls compressor oil quality for a prominent manufacturer of hermetically sealed refrigerating units. At the right—in the same plant—Bowser dehydration of sealing and lubricating oil enables a battery of vacuum pumps to pull down to 10 microns.



This exclusive Bowser method completely degasifies and dehydrates refrigerator oils as they are used at the charging boards. Oil, not previously dehydrated, may be used at lower cost per gallon.

Continuous dehydration of vacuum pump oil with a Bowser unit assures micronic range vacuums without oil changes. The addition of make-up oil may be necessary at infrequent intervals.

When *extra dry* oil or other liquids are needed for production equipment, processing operations or finished products—Bowser dehydrators and degasifiers can be counted on for new highs in the removal of air, gases, volatiles, solvents, water and solids.



ONLY 2 SIMPLE STEPS

1. The Bowser method *atomizes oil into minute particles* to expose maximum surface area.
2. *Performs the atomizing operation in a high vacuum* to separate contaminants from the oil and to prevent their reentry.

• • •

For complete data write to
Bowser, Inc., 1358 Creighton Avenue
Fort Wayne 2, Indiana

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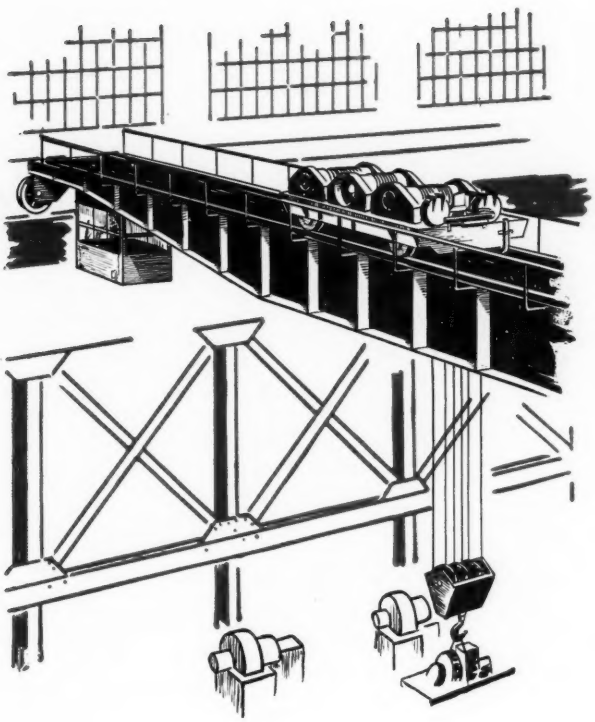
never a bearing failure with Farval



FARVAL—*Studies in Centralized Lubrication* No. 106

NINETEEN years ago, Farval Centralized Lubrication was installed on a 10-ton overhead traveling crane in one of the steel mills in the Chicago district. A recent report on this early job states that the Farval systems are still in service, as good as on the day they were installed, and that since 1930 the crane has never been down for a bearing repair or replacement. In short, there has never been a minute's trouble due to faulty lubrication.

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More than 2000 Farval systems have been installed on cranes. Many of them are roller bearing cranes, where clean grease is essential and over-lubrication should be avoided.

Farval has proven itself in over 20 years of service. It is the original Dualine system of centralized lubrication that others imitate. The Farval valve has only 2 moving parts—is simple, sure and foolproof, without springs, ball-checks or pinhole ports to cause trouble. Through its wide valve ports and full hydraulic operation, Farval unfailingly delivers grease or oil to each bearing—as much as you want, exactly measured—as often as desired. Indicators at every bearing show that each valve has functioned. For a full description, write for Bulletin No. 25.

The Farval Corp., 3267 E. 80th St., Cleveland 4, O.

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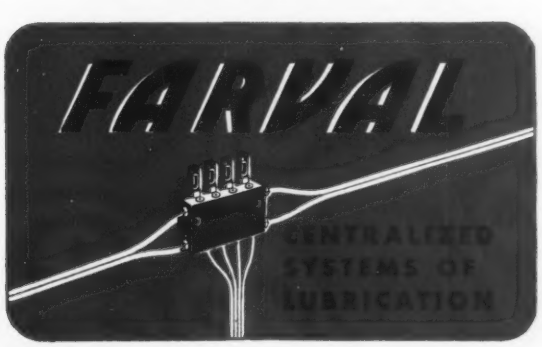

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